IN TWO SECTIONS—SECTION TWO

TRANSACTIONS

of The American Society of Mechanical Engineers

SOCIETY RECORDS—Part II

Memorial Notices of Deceased Members

Part I of Society Records for the year 1936, containing Council and Committee Personnel and other general information, was issued as Section Two of the Transactions for February, 1936

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Memorial Notices

THE purpose of Memorial Notices is to place on permanent record the biographical and professional data relating to deceased members of The American Society of Mechanical Engineers. Hence every effort is expended to insure accuracy, and to make the notices as inclusive as is reasonably possible.

The first source of information upon which these notices are based is the Society's file of membership applications and transfers. In the case of the more recent member, these application records are fairly complete. The applications of those who became members many years ago, however, contain less detailed data, and in many cases the sponsors are themselves no longer alive, so that it is difficult to obtain assistance from this source. If the member has been retired for several years prior to his death, his business associates are frequently hard to locate, and, in some cases, members of his family cannot be found. While all these factors add to the difficulty of obtaining accurate and fairly complete data, every possible source of information is explored, with the result that publication of the notice is sometimes delayed.

During the past year the Committee on Publications has, in the case of many deceased members, asked former friends and associates to prepare the obituary. The object is to secure a final record that will be more valuable for having been prepared by men who knew the deceased and are competent to evaluate his work. Memorials prepared in accordance with this recent policy are signed by those who wrote them or who collaborated in their preparation. To all persons who have thus cooperated, the Committee acknowledges its gratitude.

The Committee has also established most helpful and cordial contacts with other societies in the preparation of these notices. As members of this Society are sometimes members of others also, collaboration in the preparation of obituaries holds the possibility of securing a more completely authoritative record. Several obituaries published with this series will be found to have been republished in full from the records of other societies, or to have been adapted from these records. To these societies, and to their members who prepared the obituaries so used, the Committee wishes to express its thanks.

The Committee also appreciates and acknowledges the assistance that has been given by relatives, business associates, and friends in the preparation of all other memorials. It also acknowledges its debt to such sources as Who's Who in Engineering, Who's Who in America, and similar publications; the Encyclopedia of American Biography, National Cyclopedia of American Biography, and New International Year Book; the technical and daily press; colleges and universities and their alumni associations; and to engineering and other societies which have supplied information from their records.

Relatives, business associates, and Local Section and Student Branch officers are urged to notify the Society promptly of the deaths of members. Newspaper clippings or obituaries in any other form should be sent whenever available and the names and addresses of those who can supply further information should be furnished. A special form for supplying complete details will be forwarded by the Office of the Society upon request.

THE COMMITTEE ON PUBLICATIONS

Memorial Notices

CARL RICHARD AHLQUIST (1897–1935)

Carl Richard Ahlquist died in his native city of Denver, Colo., on June 3, 1935, of pneumonia. He was born on February 7, 1897, son of Charles Augustus and Caroline (Ditloff) Ahlquist. He attended the grade schools and the Manual Training High School in Denver and subsequently took special courses at the School of Commerce of the University of Denver.

Early in 1918 Mr. Ahlquist entered the employ of the Gates Rubber Company, Denver, in the capacity of architectural and mechanical draftsman. A few years later he was promoted to the position of chief engineer, with supervision over all construction, equipment, operation, and maintenance. He remained with the company until July 1, 1934, and was not employed at the time of his death.

Mr. Ahlquist became an associate-member of the A.S.M.E. in 1930. He was secretary of the Denver Exchange Club from 1931 to 1933 and was a Royal Arch Mason. He was gifted in painting and crayon work, and also in clay modeling, and was especially fond of flowers and of hunting and fishing. Surviving him is his widow, Ethel Lorena (Weaver) Ahlquist, whom he married in 1923.

ROBERT ANGUS (1842–1935)

Robert Angus, of London, Ontario, Canada, died on January 16, 1935, at the advanced age of 93 years. He was born at Kingston, Ontario, Canada, on January 11, 1842, of Scottish parentage, and moved with his parents to the vicinity of London when he was less than ten years old.

He had little opportunity for systematic schooling in that district, but having been apprenticed to the Honorable Elijah Leonard, a builder of sawing machines, threshing outfits, and water turbines, he entered the machinist's trade and began a long course of reading and private study by which he acquired a valuable technical knowledge. This firm later built steam engines, and Mr. Angus took a keen interest in steam engineering until within a few months of his death.

In 1874 he made the complete designs and drawings for a steam pumping engine for the St. Thomas, Ontario, waterworks, and had charge of the building and erection of this machine. The following year he designed and superintended the building and erection of a pumping engine for Sarnia, Ontario, and both machines were most successful. While with the Leonard firm, he designed complete lines of steam engines and boilers and other equipment.

When, in 1880, E. Leonard & Sons acquired from F. H. Ball the rights to build the Ball engine and governor in Canada, Mr. Angus designed the engines with which the special governor was used, and made many improvements in this type of machinery, acquiring great skill in the operation and adjustment of the governor. The latter was very sensitive and each type used required very nice adjustment of the moving parts to make the engine govern properly.

For several years, after 1888, he was master mechanic of the Standard Oil Company in Cleveland, Ohio, and during that time he was responsible for the design and building of a great deal of new equipment, just then being developed by the company for the purifying of Lima oil.

Up to within about ten years of his death he continued a consulting practice and kept himself remarkably up-to-date on engineering and other matters.

He joined the A.S.M.E. in 1891, being the first Canadian member west of Montreal. He was also a member of the Engineering Institute of Canada. His two sons, Professor Robert W. Angus, University of Toronto, and H. H. Angus, consulting engineer, Toronto, have been members of the A.S.M.E. for many years.—[Memorial prepared by Robert W. Angus, Toronto, Ontario, Canada. Past Vice-President, A.S.M.E.]

WILLIAM WALLACE ATTERBURY (1866–1935)

Many technically trained mechanical engineers, after extensive experience in railroad service, have become associated with allied industries and have advanced to high and important executive positions. Very few, however, have become chief executives of railroads. The late William Wallace Atterbury, an honorary member of the A.S.M.E., is the most outstanding example of such advancement.

What qualities, beyond that of a technical training, equipped him to become one of the great railroad executives of his generation—not alone as the chief executive of one of the world's largest and most important railway systems in an exceedingly difficult period in American railroad development, but also as director general of transportation of the American Expeditionary Forces in France, a position in which he rendered invaluable service during the World War?

A satisfactory answer to this question would require an extended biographical treatise, comparable with those which have been published by the Society in recent years, and it is sincerely to be hoped that ways and means may be found for publishing such a biography. I shall attempt here to designate only a few of the more prominent characteristics which accounted for General Atterbury's unusual success.

William Wallace Atterbury was born at New Albany, Ind., on January 31, 1866, the son of John G. and Catherine (Larned) Atterbury, and spent his early life in Detroit, Mich. He was graduated from Yale University in 1886 and shortly thereafter, on October 11, became an apprentice on the Pennsylvania Railroad, at Altoona, Pa. An apprenticeship in those days was no sinecure; the pay was five, seven, and nine cents an hour for the first three years. The working hours were 7:00 a.m. to 12 noon; 12:30 p.m. to 6:00 p.m., except on Saturday, when the shops shut down at 5:00 p.m. When business was rushing the shops were also open three evenings a week, from 7:00 p.m. to 1:00 a.m. Hard work; but happy days, according to the reports of Atterbury's associates in those early days.

James Millikan, who preceded Atterbury as an apprentice at Altoona by a few months, looked him up shortly after he reported and asked him where he lived. "I am rooming with Old Tom" (the company night policeman, and a fine, loyal fellow, who weighed more than 300 lb), said Atterbury. "He sleeps in the bed in the daytime and I sleep there at night." And then, according to Millikan, "he threw back his head and laughed that gay, contagious laugh for which he was noted all his life—the laugh that brought him many friends and carried him over many difficulties."

At the end of his apprenticeship in 1889, Atterbury was sent to Philadelphia and then to Derry, on the Pittsburgh division, as assistant road foreman of engines. There he came under J. K. Russell, a sterling, loyal, practical, old-time railroader—a just and stern disciplinarian. Atterbury, in later life when dealing with difficult labor problems, frequently acknowledged his indebtedness to Russell, who taught him the true value of organization and discipline, with fairness at all times, justice and discipline being tempered with mercy.

Atterbury served as assistant road foreman of engines on other divisions and in 1892 was promoted to assistant engineer of motive power of the Pennsylvania's Northwest System. In 1893 he was made master mechanic at Fort Wayne, Ind. Here his vigorous and successful handling of the strike of enginemen and trainmen in 1894 brought his abilities as an executive to the attention of those in authority, and on October 26, 1896, he was made superintendent of motive power of the Pennsylvania Lines East of Pittsburgh and Erie. Young college graduates, serving as special apprentices at Altoona during these years, recall with appreciation and enthusiasm the invitations for Sunday dinners in the Atterbury home. He was advanced to general superintendent of motive power on October 1, 1901.

Always of an inquiring mind and ambitious to get ahead, Atterbury took advantage of every opportunity to study operating methods and practices. He assumed all the authority his superiors or the operating officers with whom he was associated would delegate or entrust to him. The knowledge and experience thus gained proved a valuable asset during the boom in 1902, when the railroad was swamped with business and encountered one of its most serious freight congestions. Atterbury, although a mechanical-department officer and only 37 years of age, showed such a clear understanding of the complicated factors involved that President Cassatt, to the amazement of his associates, advanced him on January 1, 1903, to the general managership of the Pennsylvania Lines East of Pittsburgh and Erie.

The president's judgment was quickly vindicated, for Atterbury promptly unraveled the traffic snarl and got things running smoothly. On March 24, 1909, he was made fifth vice-president in charge of transportation. A change in organization was effected on March 3, 1911, and he was elected fourth vice-president and also a director of the company. The numerical designation of vice-presidents was discontinued on May 8, 1912, and his title was changed to vice-president in charge of operation. Then followed eventful years.

As head of the American Railway Association, to whose presidency he was elected on May 17, 1916, Atterbury rendered invaluable service in the transportation of troops and war supplies to the Mexican border, and of supplies to the eastern seaboard. It is not strange, therefore, that when General Pershing cabled in July, 1917, "Successful handling our railroad lines so important that ablest man in country

should be selected," Atterbury was sent to France and made director general of transportation for the American Expeditionary Forces. It was a stupendous task, in a foreign land and under the stress of the Great War, to organize to meet the United States transportation requirements in France and to harmonize the facilities and efforts with those of our Allies. He sailed for Europe in August, 1917, and was commissioned Brigadier-General of the United States Army, October 5, 1917. He returned to America on May 31, 1919, to resume his duties on the Pennsylvania Railroad.

That his efforts in the World War were eminently satisfactory is indicated by the many decorations which he received, including the Distinguished Service Medal from the United States; the Legion of Honor, Rank of Commander, from France; Companion of the Most Honorable Order of the Bath, from Great Britain; Commander of the Order of the Crown, from Belgium; Royal Order of the White Eagle, from Serbia; and Grand Officer of the Order of the Crown, from Rumania.

Postwar conditions on the railroads, with the trying problems of rehabilitation and readjustment and severe labor difficulties, were hardly less serious and difficult than those involved in the War, and General Atterbury, as head of the operating department of the Pennsylvania Railroad, exercised an influence and leadership which had a profound effect upon the progress and improvement of the American railway system. With courage and directness he approached the problem of discovering a successful method of settling differences arising between the management and the employees. This task on a railroad, with its different types of labor organizations, including the brotherhoods, unions affiliated with the American Federation of Labor, and other types, is much more involved and difficult than in other industries. The situation is further complicated by the fact that governmental agencies enter the situation more actively, the railroads being public-service institutions.

General Atterbury's success in dealing with his coworkers may be gaged by the following statement of one of the labor leaders. "I cannot help but feel that had General Atterbury the choice, he would rather be known to posterity as the originator of the plan which brought permanent peace and goodwill to the Pennsylvania Railroad and its employees, than as the great railroad executive that he was."

Shrewd labor leaders, sitting across the table from representatives of management, appraise their opponents with keen and ruthless discrimination. Here is the appraisal of a labor leader who had contacts with General Atterbury over a quarter of a century, or more. "Those phases of General Atterbury's personality with which I have been most deeply impressed were his unconscious naturalness, his directness of approach to the subject under consideration, his rigid adherence to his promises or decisions once they were given (there was no splitting of hairs with General Atterbury when it came to applying any agreement or understanding he had made with the Pennsylvania employees), and his consideration of matters, not on the basis that they must be settled in a certain manner because he, the one in authority, willed it so, but on the basis of what should be done, giving proper regard to the rights and interests of all concerned."

The public at large frequently has peculiar conceptions of the lives of men of prominence. The real characters of these men are best known and understood by their intimates who work with them "behind the scenes."

"A notable trait of the General," said one of his intimate coworkers, both on the Pennsylvania Railroad and in France, "was his courage, poise, and patience in the face of difficulties."

"In any matter of moment," said this same associate, "once he reached a definite conclusion as to the course which should be pursued, he proceeded accordingly and calmly awaited developments. His conclusions were reached after careful consideration of advice and suggestions, which he at all times welcomed from any source. Whenever he was confronted with unusual difficulties he had a way of doing all he could in the circumstances, then relaxing, and tackling the next day's work in the light of conditions as they unfolded.

"Incidentally, the ability to throw off cares and relax was a prominent attribute of the General. He was a splendid example of the type of executive who knows how to blend work and recreation and who functions in an orderly fashion without stress or strain. Golf, cards, and light reading were his favorite forms of recreation."

General Atterbury was elected vice-president on November 15, 1924, and advanced to the presidency on October 1, 1925. One of his last great achievements is summed up in the following paragraph, which is taken from a minute adopted by the board of directors of the Pennsylvania Railroad Company at its meeting on April 24, 1935, when, because of ill health, he declined to accept nomination for reelection as president.

"His courage in the face of obstacles is well exemplified by his prosecution of the work of electrifying the lines of the company between New York and Washington. Begun in 1928, the work was halted by the difficulty of financing, brought about by the worldwide depression. Upon the offer of the Government to advance funds, Mr. Atterbury did not hesitate; he accepted the offer and pushed the work to a conclusion, so that the lines were open for operation in February of this year."

It was at his suggestion, also, that the Association of American Railroads was devised to promote and improve railroad service in the public interest and to maintain the integrity and credit of the industry. This association, organized on October 12, 1934, makes it possible for the railroads to present a united front in problems which face the industry as a whole.

General Atterbury was among the first of those leaders who have a vision of a coordinated transportation system for this country—a system in which each type of transportation will take that place for which it is best suited on a sound economic basis, and in the public interest. Although, from the standpoint of practical accomplishment, comparatively little has been done up to the present time, General Atterbury helped to lay the foundation upon which substantial progress can be confidently anticipated in the years immediately before us.

General Atterbury died on September 20, 1935, following a long illness. He was a member of many learned societies. Honorary membership in The American Society of Mechanical Engineers was conferred upon him at a memorable meeting in Altoona, Pa., in 1925. He had been a member of the Society since 1894. He was given the honorary degree of master of arts by Yale University in 1911. Several universities conferred upon him the degree of doctor of laws, including the University of Pennsylvania in 1919, Yale University in 1925, Villanova College in 1927, and Temple University in 1929. Pennsylvania Military College gave him the degree of doctor of engineering in 1932.

General Atterbury married twice. Five years after the death of the first Mrs. Atterbury, the former Miss Minnie H. Hoffman, of Fort Wayne, Ind., he was married in 1915 to Mrs. Arminia McLeod, daughter of Mr. and Mrs. H. B. Rosengarten, of Philadelphia. He is survived by her and by a son, W. W. Atterbury, Jr.; also by three children of the second Mrs. Atterbury before her marriage to him: George R. Atterbury, Malcolm Atterbury, and Mrs. Elizabeth (Atterbury) Connelly, of Radnor, Pa.—[Memorial prepared by Roy V. Wright, New York, N.Y. Past-President, A.S.M.E.]

PHILIP DeCATESBY BALL (1864–1933)

Philip DeCatesby Ball died at St. John's Hospital, St. Louis, Mo., on October 22, 1933. One of his associates in the St. Louis Section of the A.S.M.E., Victor J. Azbe, wrote of him, shortly after his death:

"There are indeed few men born who distribute the time of their life's span as productively and as widely as did Philip Ball. From the creation of the 1000-ton cross-compound Corliss refrigerating machine—'the largest in the world'—featured at the St. Louis World's Fair, his interest ranged to the lightest of aviation engines and airplanes; and again, from engineering to sports.

"He was a worker, a builder, a leader, untiring in his efforts—untiring to such an extent that contrary to the orders of his physicians and the entreaties of his family and friends, he carried on his duties even from the hospital.

"A sportsman to the innermost core, he would absorb the annual deficit of his baseball teams unflinchingly. He carried on when a loss was certain, entirely for the sake of sport. As an airplane enthusiast, he refused to consider any other form of long-distance travel.

"He was one of the most valuable members of the St. Louis Section of the A.S.M.E."

Mr. Ball became a member of the A.S.M.E. in 1899 and gave liberally, both of time and money, to the work of the St. Louis Section. He was largely responsible for the establishment of the Spirit of St. Louis Medal, endowed by members of the A.S.M.E. and citizens residing in St. Louis, to be awarded for meritorious service in aeronautical engineering.

Born at Keokuk, Iowa, on October 22, 1864, the son of Captain Charles J. Ball and Caroline (Paulison) Ball, he was educated largely by his father, although he attended public and private schools. At the age of sixteen he became an apprentice in the shop and drafting room of his father, who was a manufacturer of ice-making machinery. He bought the business when he became of age and developed it greatly in the years that followed. The machine to which Mr. Azbe refers, exhibited at the Fair in 1904, attracted wide attention to its builder. It was sold to the Anheuser-Busch Brewing Association.

He also built many icing stations for railroads throughout the country, and operated them as private enterprises. He built the Federal Cold Storage Company plant in St. Louis, and was owner of the Mound City Ice & Cold Storage Co. and the Winter Garden & Ice Co., also of St. Louis. In 1926 he sold his ice-making machinery plant

and affiliated interests to the City Ice & Fuel Company of Cleveland but continued to take an active part in the management of that company until his death.

Mr. Ball's other interests, in addition to aviation and baseball, included drilling oil wells, colonization of land in Texas, insurance, and silver fox farms.

He bought the Ryan Aircraft Corporation, which built Colonel Lindbergh's "Spirit of St. Louis," later selling the corporation to the Detroit Aircraft Corporation. He then purchased the Monocoupe Corporation, which he operated until his death. He acquired an airplane for his own use in 1927, replacing it with later types as new designs were developed. He financed the development of a 1000-hp Diesel engine for aircraft, designed by D. J. Deschamps.

His interest in baseball led him to become one of the organizers of the Federal League of Professional Baseball Clubs, and when that was dissolved he acquired the St. Louis Browns, in 1916. Aviation and baseball were his chief interests during his last ten years.

Mr. Ball was also a member of the American Society of Refrigerating Engineers, American Society of Military Engineers, Engineers' Club of St. Louis, and the Masonic fraternity, in which he was a Royal Arch Mason and a member of the Scottish Rite. His clubs included the Athletic and Mid-Day in Chicago, the Missouri Athletic, the Noonday and Racquet clubs in St. Louis, and several country clubs in that vicinity.

Mr. Ball married Miss Harriet R. Heiskell, of Indianapolis, Ind., in 1885 and is survived by her and by two children, Mrs. Margaret (William R.) Cady and James P. Ball.

JOHN BATH (1864–1935)

John Bath, president and treasurer of John Bath & Co., Inc., Worcester, Mass., died at Shrewsbury, Mass., of a cerebral hemorrhage on July 9, 1935. He was born at Frostburg, Md., on April 12, 1864, and when twelve years old went to work in the mines. After a short time, however, he began an apprenticeship at the carshops at Huntington, Pa., later transferring to shops at Readville, Mass. After completing his apprenticeship he obtained employment with the Becker Brainard Milling Machine Company, Hyde Park, Mass., where he worked for several years, becoming thoroughly trained as a machinist. While there he invented the Bath indicator, the first instrument of its kind to measure in thousandths of an inch.

He was next connected with the Norton Company, Worcester, Mass., and subsequently developed a universal cutter grinder for the Waltham Watch Tool Company. He then founded the Bath Grinder Company, in Fitchburg, Mass., where he developed the Bath universal and plain grinders. About the year 1912, he established the firm of John Bath & Co., Inc., where for many years he engaged in designing and manufacturing special machinery, jigs, fixtures, fine gages, and tools. During the World War the plant was devoted to the manufacture of special tools, gages, and jigs used in producing gun carriages, gun parts, aircraft, and other ordnance equipment.

Of this later work and other inventions an obituary published in *American Machinist*, July 31, 1935, page 560c, says:

"After the War Mr. Bath concluded that his efforts could best be expended in developing more accurate taps. His experiments resulted in the production of taps made of high-speed steel, with the entire thread ground from the solid after hardening.

"The automotive industry was quick to recognize the savings this tap afforded, since it solved the problem of duplicating size. The demand spread to other industries, and today the high-speed ground thread tap is standard where production and accuracy are necessary.

"Mr. Bath was also the inventor of the Bath internal micrometer—an instrument for measuring internal diameters up to 0.001 in. He was widely recognized as a pioneer in precision thread production."

Mr. Bath became a member of the A.S.M.E. in 1918. He was an active member of the Sectional Committee on Standardization of Plain Limit Gages for General Engineering Work, having served as a member at large on that committee from the time of its organization in 1920 until its reorganization in 1930. This committee then became the Sectional Committee on Standardization of Allowances and Tolerances for Cylindrical Parts and Limit Gages and Mr. Bath accepted membership thereon from the time of its inception. In this capacity he represented the Drill and Reamer Society, the Milling Cutter Society, and the Tap and Die Institute on this sectional committee. He continued in this work until 1932.

Mr. Bath belonged to the Worcester Country Club and to the Rotary Club in that city, and had attained high rank in the Masonic fraternity. His wife, Carrie M. (Whitney) Bath, whom he married in 1887, died in February, 1935. He is survived by a daughter, Mrs. Jessie E. (Bath) Newcombe, of Fitchburg, and by three sons, J. Chester Bath, of Worcester, and Stanley W. and Russell F. Bath, of Shrewsbury.

WILLIAM BAYLEY (1845–1934)

William Bayley, president and chief engineer of The William Bayley Company, Springfield, Ohio, died in that city on February 4, 1934. Born in Baltimore, Md., on July 28, 1845, he was the youngest by twenty years of a family of eleven children. His mother, Mary Ann (Mason) Bayley, a native of Boston, was of English parentage, and his father, William Bayley, came to the United States from England in 1812, at the age of fourteen.

Mr. Bayley was educated in Baltimore, attending Knapp's Academy, Horshaw Academy, and the Maryland Institute, where he subsequently engaged in teaching, before he was twenty years old. In 1862 he became an apprentice machinist with Poole & Hunt (now the Poole Foundry & Machine Co.), Baltimore, where he secured the experience necessary for machine design and served the company as chief draftsman for five years.

From 1870 to 1872 he was engaged in drafting and designing for Pusey, Jones & Co. (now the Pusey & Jones Corp.), shipbuilders and machinery manufacturers of Wilmington, Del. In 1871 he patented his water wheel, a notable feature of which was its diverging discharge buckets, now in universal use. An arrangement was made whereby the Pusey & Jones Co. manufactured these wheels on a royalty basis. In 1872 Mr. Bayley organized the firm of Remington, Bayley & Co., Wilmington, to produce the wheels. It also made automatic nut punches, steam engines, and paper and sawmill machinery. He sold his interest in 1875, went to Springfield, and entered the employ of Whitely, Fassler & Kelly, manufacturers of agricultural machinery, which later became The Wm. N. Whitely. Mr. Bayley was in charge of drafting-room and experimental work until 1889, when the company went out of business.

When Mr. Bayley entered the agricultural machinery business, the automatic binder for grain was a new development. Wire was being used to secure the grain in bundles and farmers were receiving many complaints from millers because slivers of wire in the threshed grain played havoc with the bolting cloth. Whitely, Fassler & Kelly purchased the patents of Appleby, a job machine shop man of Wisconsin, who had devised a practical twine knotter, and Mr. Bayley made such adaptations and improvements as were necessary to make the "knotter" satisfactory for various kinds and conditions of grain, kinds of string, variations in weather, and other conditions affecting its use. During the period of his employment by the company, his name appears continuously in the patent records in connection with the "knotter" and many other devices.

In 1889, Mr. Bayley became part owner of The Rogers Iron Company, which had been started in 1884. This later was reorganized as The William Bayley Company, of which he was president and chief engineer until his death. In recent years he had developed the first machinery for automatic paving of streets and roads with blocks and also with concrete, asphalt, etc.

Mr. Bayley became a member of the A.S.M.E. in 1904. He belonged to the Scottish Rite in the Masonic fraternity and was a Shriner.

Surviving Mr. Bayley are four sons, William D., Guy D., Lee, and Elden Bayley; a grandson, Joe Pratt, of Chapel Hill, N.C., son of an only daughter; and ten other grandchildren. His wife Mary (Dicus) Bayley, predeceased him by six weeks.

ROBERT HERBERT BEAUMONT (1873–1933)

Robert Herbert Beaumont, president of R. H. Beaumont Co., Philadelphia, Pa., whose death occurred on April 23, 1933, was born at Woodville, Miss., on March 2, 1873. His parents were Edward Herbert and Betsy (Schofield) Beaumont. He attended the Woodville public schools and pursued various subjects in home-study courses.

At the age of nineteen he began work as a blueprint boy for the Link-Belt Company, in which he worked up to the position of salesman. During his connection with the Link-Belt Company he designed considerable machinery for coal, coke, and ash-handling systems and in 1905 he established his own contracting firm, specializing exclusively in complete plants for handling coal, coke, and ashes for boiler and gas houses, including the steel and concrete bunkers and structures. Among the specialties designed and built by the company were the Beaumont skip hoists for coal, coke, and ashes, and the Beaumont cable drag scraper system for coal storage. At the time of his death he was serving as chairman of the board of R. H. Beaumont Co. (now the Beaumont Birch Company) and of the Beaumont Manufacturing Company.

Mr. Beaumont joined the A.S.M.E. as an associate early in 1921 and was promoted to full membership later the same year.

His interests outside his business included motion pictures, building radio sets, and, particularly, aviation. He was one of the founders, in 1909, of the Aero Club of Pennsylvania and was its first vice-

president. In later years he had a plane of his own and did considerable flying in the vicinity of Philadelphia. Mr. Beaumont was twice married. He is survived by a son by his first marriage and by his widow, Mabel (Parker) Beaumont, and their four daughters.

DAVID BELL (1875–1934)

David Bell, president and treasurer of the David Bell Co., Inc., Buffalo, N.Y., manufacturers of screw-machine products, died in that city on May 21, 1934. He was born in Buffalo on September 3, 1875, the son of David and Jane (Adams) Bell, and secured his education in private and public schools there. For some years he was associated with his father in the David Bell Engineering Works, builders of marine engines, steel boats, steam hammers, and general machinery. After the incorporation of the company about 1892 he became general manager and vice-president and served in those capacities until the consolidation of the company with the Buffalo Foundry & Machine Co. in 1907. He was made general engineer, in charge of the machine shop department of the combined firm, and held that position until he became president and treasurer of the Bell company in 1918. He had taken out numerous patents on steam hammers.

Mr. Bell became a member of the A.S.M.E. in 1912. He served as secretary of the Engineering Society of Buffalo in 1913–1914, president in 1914–1915, director in 1915–1918, and vice-president, 1921–1922. He was formerly president of the Western Division of the Screw Machine Products Association, and after the National Screw Machine Products Association was formed in 1932 served as its president until his death.

Just prior to his death Mr. Bell served for three years as president of the Board of Trustees of the Lafayette Avenue Presbyterian Church, Buffalo. He was a member of the Buffalo Athletic Club and of the Park Club and the Meadowbrook Country Club. He is survived by his widow, Alice Willard (Nash) Bell, whom he married in 1901, and by a daughter, Mrs. Charles H. Hyde, of Buffalo.

JOSEPH PROSPER BERANGER (1890–1935)

At the time of his death, Joseph Prosper Beranger was vice-president of the West Indies Sugar Corporation in charge of its operations in Santo Domingo. For a quarter of a century he was an active figure in the sugar industry of the Caribbean countries, rising through the merits of his services to executive positions of great responsibility and trust.

Mr. Beranger was born at New Orleans, La., on March 31, 1890, a son of Joseph J. and Eveline (Flouse) Beranger and a descendant of a French family established at New Orleans in 1842 by his grandfather, J. Justin Beranger. Mr. Beranger's father, a retired ship builder, is still living in that city.

Joseph Prosper Beranger received his preliminary education in the public schools of New Orleans and at Rugby Academy in that city and subsequently entered Tulane University, from which he was graduated in 1911 with a bachelor's degree in engineering. His interests already centered in the sugar industry, and in the latter part of June, 1911, he secured a position as junior engineer at the Central Guanico Sugar plantation, in Puerto Rico, of the South Puerto Rico Sugar Company. Here he remained until September, 1912. Returning to Louisiana at the end of that time, he entered Louisiana State University at Baton Rouge for postgraduate work in engineering. He took the degree of master of science in 1913 and subsequently was assistant to Professor E. W. Kerr (Mem. A.S.M.E.) in experimental and testing work at sugar houses throughout Louisiana for the Louisiana State Experimental Station. He was next engaged in designing and drafting at the Central Fortuna, Ponce, Puerto Rico.

In the fall of 1914 he was appointed assistant engineer of the West India Management & Consultation Co. While serving as such, he was in charge of the rebuilding of the Central Ansonia sugar factory in Santo Domingo (1914) and also of the rebuilding of the Central Quisqueya Sugar factory (1915). In June, 1915, in behalf of the West India Management & Consultation Co., he became resident supervising engineer in charge of the construction of the Central Punta Alegre at Caibarien, Cuba, which was the largest sugar mill built up to that time.

In June, 1917, Mr. Beranger was promoted to the post of chief engineer of the West India Management & Consultation Co., with headquarters in New York. During the following year, the West India Sugar Finance Corporation was formed to finance sugar centrals in Puerto Rico, Santo Domingo, and Cuba. This corporation took over control of the West India Management & Consultation Co. and retained Mr. Beranger as chief engineer, in which capacity he designed and erected the sugar factory at Barahona, Santo Domingo. He also designed and was in charge of the erection of a sugar factory at Central Tanamo, Cuba.

From 1920 to 1922, Mr. Beranger continued as chief engineer and purchasing agent for the properties financed, owned, or controlled by the West India Sugar Finance Corporation in Puerto Rico, Cuba, and Santo Domingo, consisting of Fajardo Sugar and Central Corsica in Puerto Rico; Centrals Palma, Alto Cedro, America, Cupey, Santa Ana, Hatillo, and Altagracia, all in Cuba; and Centrals Consuelo, San Isidro, Quisqueya, Las Pajas, Porvenir, and Ansonia in Santo Domingo.

In 1922, the Cuban Dominican Sugar Company was incorporated to purchase those of the aforementioned properties owned or controlled by the West India Sugar Finance Corporation, and Mr. Beranger was retained in the capacity of chief engineer and supervisor of production. In 1926, the Cuban Dominican Sugar Corporation was incorporated as the successor of the Cuban Dominican Sugar Company and Mr. Beranger was appointed assistant to the president, George H. Houston (Mem. A.S.M.E.). In this capacity he continued to supervise all of the factory operations and production of the properties then owned by the corporation, namely, Centrals Cupey, Alto Cedro, Palma, America, Santa Ana, Hatillo, and Altagracia in Cuba; Centrals Consuelo, Quisqueya, San Isidro, and Las Pajas in Santo Domingo.

In 1927, Mr. Houston resigned, and Frederick B. Adams succeeded him as president of Cuban Dominican Sugar Corporation. Mr. Beranger was installed as vice-president in charge of the operations in Cuba and Santo Domingo of the various properties. He continued in this capacity until late in 1928, when the importance of the sugar properties in Santo Domingo, which included a large irrigation project located at Barahona, made it necessary for him to discontinue supervision of the Cuban sugar estates and devote his entire time to those in Santo Domingo, of which he was in charge until his death. In 1921, he was also elected vice-president of the Cuban Dominican Sales Corporation, a subsidiary of the Cuban Dominican Sugar Corporation, in charge of selling the sugar produced by the Cuban Dominican properties as well as the purchase of supplies for these estates.

Early in 1931, due to the inability of the Cuban Dominican Sugar Corporation to pay interest on its bonds, a plan of reorganization was prepared and eventually adopted by the security holders. This resulted in foreclosure sale of the properties in November, 1932, and their purchase by a new corporation, the West Indies Sugar Corporation. Mr. Beranger was appointed vice-president in charge of operations in Santo Domingo, comprising supervision over Centrals Barahona, Consuelo, Quisqueya, Las Pajas, and San Isidro. In 1933, he was elected a director of the Compania Azucarera Dominicana, the subsidiary in Santo Domingo which owned all of these properties except Central Barahona. He was a gifted executive and engineer who possessed an exceptional knowledge of the industry to which he devoted his life, and contributed substantially to the progress of the organizations with which he was connected. Mr. Beranger did a great deal of creative work in connection with the development of the various plantations. In particular, he was largely responsible for the growth of Central Barahona, which is located in desert lands, and which, from the standpoint of operation, is considered one of the most outstanding plantations in the world. Mr. Beranger was of the opinion that the cost of producing sugar at Barahona was lower than that of any other Central.

Mr. Beranger became a junior member of the A.S.M.E. in 1914, an associate-member in 1917, and a member in 1926. He was an original director of the Dominican Chamber of Commerce, a member of the Pan American Society, a former member of the Havana Country Club, and a member of the New York Produce Exchange Luncheon Club, the Engineers' Club of New York, the Columbia Yacht Club, and the Arcola Country Club, of New Jersey. During the World War he was in Washington, where his knowledge of the sugar industry was at the disposal of the Government and was drawn upon in shaping wartime plans. Mr. Beranger was very much interested in the Salvation Army and in such boys' organizations as the Boy Scouts of America and the Madison Boys' Club of New York, which he supported. He was always fond of outdoor life and sports, playing on the tennis, baseball, and football teams in his college days, and in later years finding his principal recreation in golf.

Mr. Beranger married Mrs. Vida (French) Barry, daughter of Harry Delmar French, founder of the Brooklyn School of Journalism, in 1928, and is survived by her and by her daughter by an earlier marriage, Barbara French Beranger, who adopted her stepfather's name.

Mr. Beranger died at Miami, Fla., on March 26, 1935, a few days before his forty-fifth birthday. In spite of his long years of service in the sugar industry and his many accomplishments, he was still a relatively young man and the abrupt termination of his career was a serious loss to the organizations with which he was connected.

He was known and admired throughout the industry and enjoyed the warm personal regard of all with whom he was associated.—[From the "Encyclopedia of American Biography," a compilation of The American Historical Society, Inc.]

HENRY E. BILGER (1865–1935)

Henry E. Bilger, who terminated his work at the Baldwin Locomotive Works in February, 1932, after having been connected with the small tool and gage department for fifty years, died on April 23, 1935, following a brief illness. Mr. Bilger was born at Philadelphia, Pa., on March 8, 1865, son of Samuel Ellsworth and Frances Sophia (Johnson) Bilger. After his graduation from public school he was apprenticed to the Baldwin Locomotive Works to learn the trade of a machinist. Following a four-year apprenticeship he worked as a journeyman machinist until July, 1889, when he was made foreman of the tool department. In 1915 he became superintendent of tools and gages in the Eddystone Ammunition Corporation, but returned to the Baldwin Works early the following year to supervise French commission work. He was then made superintendent of machine tools and machine tool repairs. Mr. Bilger designed and manufactured tools, fixtures, and gages for the construction of steam, electric, and gasoline locomotives, built many models, and conducted numerous experiments. During the World War he directed a large volume of work on tools and gages for Russian 3-in. shrapnel, 4.7-, 6-, and 12-in. high-explosive English shells, and 220- and 270-mm. French shells. He contributed many articles to *American Machinist.*

Mr. Bilger became a member of the A.S.M.E. in 1916 and was a Royal Arch Mason. He was superintendent of the Sunday School of the Spring Garden Methodist Episcopal Church, Philadelphia, for twenty years, from 1899 to 1919. He liked to sketch and was interested in stamp collecting.

Surviving Mr. Bilger are his widow, Florence E. (Brenniser) Bilger, whom he married in 1903, and two daughters, Louise W. Bilger and Elizabeth (Bilger) Stevens.

CHARLES FABEN BLEYER (1883–1935)

Charles Faben Bleyer, superintendent of the Fuels and Power Department of the National Tube Company, Lorain, Ohio, died suddenly on January 15, 1935, at his home in that city. He is survived by his widow, Clara (Engelfried) Bleyer, whom he married in 1909, and by a daughter, Constance Bleyer.

Son of Julius and Elizabeth (Faben) Bleyer, he was born on January 21, 1883, at Milwaukee, Wis., where he secured his education, attending the grade and high schools and the University of Wisconsin. After receiving his degree in mechanical engineering in 1907 he took a position with the Allis-Chalmers Company as erecting engineer and during the next twelve years installed and tested gas engines for many plants in the company's eastern territory, including large units for the American Steel & Wire Co., the Pittsburgh Plate Glass Company, the National Tube Company, and the Goodyear Tire & Rubber Co. During 1916 he engaged in design, estimating, and sales work for the Allis-Chalmers Company, in connection with the larger size electric hoists for mines, but finding it less interesting than erection work, returned to his former position during the World War.

When Mr. Bleyer first became associated with the National Tube Company in the fall of 1919 he was put in charge of the blast-furnace power house and boiler rooms and spent much of his time on improving operating methods and increasing plant economy. He was appointed general master mechanic of blast furnaces, coke plants, and docks, in addition to his other duties, in 1920, and five years later was advanced to the position of the superintendent of the gas and steam power department. He held a patent on furnace pressure control.

Mr. Bleyer became a junior member of the A.S.M.E. in 1909 and a member in 1920. He also belonged to the Cleveland Engineering Society and to the Masonic fraternity, Odd Fellows, and Tau Beta Pi. He served on the Lorain City Engineers Committee for the improvement of the city water works in 1931, and in 1934 was a member of a committee for investigating public utility rates in Lorain.

ALFRED PANCOAST BOLLER, JR. (1869–1935)

Alfred Pancoast Boller, Jr., whose death occurred on January 14, 1935, was born in East Orange, N.J., on April 7, 1869, a son of Alfred Pancoast and Katharine (Newbold) Boller. He attended the Dearborn-Morgan School in Orange and Stevens Institute of Technology, Hoboken, N.J., from which he received a mechanical engineering degree in 1891.

Following his graduation, Mr. Boller spent about a year each in the drafting rooms and shops of Henry R. Worthington in Brooklyn. After some experience in dike building and dredging on the upper Hudson River for the International Contracting Company, New York, N.Y., he returned to the employ of Mr. Worthington, supervising construction work and tests of pumping engines. For a year he was in charge of the Worthington Water Meter Department and he spent two and a half years in the Hawaiian Islands, directing constructing, installing, and testing complete pumping plants for irrigating.

In 1902 Mr. Boller went to Pittsburgh, Pa., to work for the Westinghouse Electric & Manufacturing Co., where he remained for two years. The next two years were spent with the Chapman Ball Bearing Company, Boston, Mass. In 1906 he took up consulting practice in New York and continued in that work for eight years. From 1914 to 1931, when he retired, he was consulting engineer for the Stanley Rule & Level Co., New Britain, Conn.

Mr. Boller became a member of the A.S.M.E. in 1901. He also belonged to the Masonic fraternity. His wife, Anna (Hawley) Boller, died in 1932. A brother, Richard E. Boller, survives him.

GEORGE MEADE BOND (1852–1935)

George Meade Bond, for fifty-three years an active member of the A.S.M.E., died at his home in Hartford, Conn., on January 6, 1935, after a long illness.

He was born on July 17, 1852, at Newburyport, Mass., son of Daniel George and Wilhelmina (Kruger) Bond. His father died two years later and his mother with her two sons moved to Grand Rapids, Mich., where George attended school until he was seventeen, and then taught in neighboring schools for a few years. His tastes running to mechanics rather than teaching, he served an apprenticeship in a Grand Rapids machine shop and continued as machinist in the Phoenix Furniture factory until his savings permitted him to enter Stevens Institute of Technology in 1876, at the age of twenty-four. He was graduated in 1880 with the degree of mechanical engineer.

During his senior year, 1879, Professor James E. Denton, recognizing Mr. Bond's unusual mechanical genius, brought him into association with William A. Rogers, professor of astronomy at Harvard College Observatory, Cambridge, and so began a cooperation which oriented Mr. Bond's entire life work, directing it to the establishment of a standard system of measurement, on which our prevailing system of interchangeable manufacture so largely depends.

Working under the direction of Professor Rogers, Mr. Bond designed the Rogers-Bond comparator, which permitted the comparison of both end-measure and line-measure standards with a degree of accuracy not previously practicable. While the aim of Professor Rogers was to improve the accuracy of his astronomical work, Mr. Bond, with his shop training and background, foresaw the advantages to the mechanical trades of accurate standards and ready means of making precise measurements. He completed the drawings of the comparator during his senior year at Stevens Institute in 1880, and on July 1 of that year entered the employ of the Pratt & Whitney Co. of Hartford to direct the building of the comparator. Two of these were built, one of which is still retained by the Pratt & Whitney Co., the other going to Professor Rogers at Cambridge for his scientific work. The latter has since been transferred to the Johns Hopkins University at Baltimore.

The A.S.M.E. Committee on Standards and Gauges at the annual meeting in November, 1882, presented an elaborate report[1] on the Rogers-Bond comparator, from which we quote:

"This apparatus is used, first, to compare line measures of length with attested copies of the standard bars of England and the United States; second, to subdivide these line measures into their aliquot parts, and to investigate and determine the errors, if any, of these subdivisions; and, third, to reduce these line measures to end measures for practical use in the shops." And from the conclusion of the report: "In the opinion of this committee, the degree of accuracy already attained is such that no future improvements can occasion changes sufficiently great to affect the practical usefulness of the magnitudes here determined, or the interchangeability of structures based upon them with those involving further refinements."

In view of this remarkable tribute of fifty-two years ago to the work of Rogers and Bond, it is interesting to note the names signed to the report: J. Sellers Bancroft, secretary of the Committee; Henry Morton, president; S. W. Robinson, Oberlin Smith, E. H. Parks, Ambrose Swasey, Chas. T. Porter, Alfred Betts, George R. Stetson.

By means of the comparator Mr. Bond developed for the Pratt & Whitney Co. standard bars (line measures) having very small and known errors from the recognized international standards. The next step was the development of the Bond measuring machine, using

[1] Trans. A.S.M.E., vol. 4 (1883), pp. 21–29.

similar bars and simplified for toolroom use. Hundreds of these machines have been built and are in daily use throughout the United States and in many foreign countries.

Mr. Bond was manager of the Standards and Gauge Department of the Pratt & Whitney Co. from 1880 until 1902, during which time he developed their line of cylindrical, caliper, snap, limit, thread, and other gages. The need of uniformity in sizes and threads of bolts, nuts, and screws for railroad service was the incentive for much of this preliminary work. As early as 1864 the inconvenience and confusion resulting from diversity in screw threads in use was brought up for consideration before The Franklin Institute. In 1882 the Master Car Builders' Association, which with other railroad organizations had adopted the Sellers or Franklin Institute system of threads, asked the Pratt & Whitney Co. to prepare sets of standard screw-thread gages in order to secure uniformity of product, and Mr. Bond, again working with Professor Rogers, developed means for both making and measuring screw threads with great accuracy. This work brought the highest commendation from the Master Car Builders' Association and the Master Mechanics' Association, whose later reports testified to remarkable savings in time and cost of railroad repair work as a direct result of this standardization.

Our present systems of mass production, based upon interchangeable manufacture without the necessity of hand fitting or selective assembly, commonly referred to abroad as the "American System," owes a debt to Mr. Bond, not for its original conception, which goes back to Eli Whitney more than a hundred years ago, but for practical means of measuring with a degree of accuracy not before available. He liked to preach what he practiced, and made many notable contributions to the proceedings of various technical societies. See A.S.M.E. Transactions, vol. 2 (1881), p. 81, "Standard Measurements"; vol. 3 (1882), p. 122, "A Standard Gauge System," and vol. 7 (1886), p. 311, "Standard Pipe and Pipe Threads"; also discussions on other papers. On February 21, 1884, he delivered a lecture before The Franklin Institute on "Standards of Length and Their Subdivision," and a week later, before the same body, another lecture on "Standards of Length as Applied to Gauge Dimensions." His book entitled "Standards of Length and Their Practical Application" was published by the Pratt & Whitney Co. in 1887. This was also given in lecture form before the Society of Arts, Boston, in 1888. In 1890 he delivered a paper before the American Association for the Advancement of Science on "Internal Strains in Hardened Steel."

Mr. Bond became a member of the A.S.M.E. in 1881, and in 1931 received the Society's 50-year medal, although his last illness was already upon him and he was unable to attend the Annual Meeting. He served as Manager of the Society from 1888 to 1891 and as Vice-President from 1908 to 1910. He was a member of the Society's various pipe-thread committees, and at the time of his death was the Society's representative on the Sectional Committee on Pipe Threads. In 1902 and 1903 he was a member of committees on the metric system, and appeared for the Society before the Congressional Committee on Coinage, Weights, and Measures to oppose the proposed adoption of the metric system as compulsory.

Mr. Bond held the oldest membership in the Engineers' Club of New York, dating from 1889. He was a member of the American Society of Civil Engineers and for a time chairman of its Committee on Units of Measurement. He was a Fellow of the American Association for the Advancement of Science and a member of the New England Historic Genealogical Society, Connecticut Historical Society, Hartford Club, Union League and Transportation Clubs of New York, and of the Royal Societies Club of London, also of the Alumni Association of Stevens Institute, of which he was president, 1886–1887.

In 1921 Stevens Institute conferred on him the degree of doctor of engineering, and in 1927 Trinity College conferred the degree of master of science.

Mr. Bond never married, and left no close relatives.—[Memorial prepared by B. H. Blood, Retired, Hartford, Conn. Formerly Mem. A.S.M.E., and connected with the Pratt & Whitney Co. for many years.]

FRANKLIN MEYER BOWMAN (1870–1934)

Franklin Meyer Bowman, vice-president of the Blaw-Knox Company, Pittsburgh, Pa., died suddenly on October 12, 1934, at his home in Pittsburgh, Pa.

Mr. Bowman was born at Freeport, Ontario, Canada, on September 2, 1870. He prepared for college in schools at Berlin, Ontario, and was graduated in civil engineering from the school of Practical Science, Toronto, in April, 1890. During the following year he served an apprenticeship as a land surveyor and after passing the government examinations and securing his title in 1891 he came to the United States and was engaged as a draftsman by the Pennsylvania Steel Company at Steelton, Pa., in the Bridge and Construction Department.

In April, 1892, he became associated with the Structural Iron Works of the Riter-Conley Manufacturing Company, at Allegheny, Pa., as draftsman, and after a few months was promoted to the position of chief draftsman. This early experience enabled him to secure the degree of civil engineer from Toronto University in 1893. He remained with the Riter-Conley company until 1912, serving successively as assistant engineer, chief engineer, director, and secretary. He resigned to become vice-president of the Blaw-Knox Company. In that capacity he was in charge of the Transmission Tower and Structural Steel Divisions of the company. He was actively identified with transmission-line design and erection of international significance and the construction of some of the largest mills in the United States.

Mr. Bowman became a junior member of the A.S.M.E. in 1894 and a member in 1902. He also belonged to the American Iron and Steel Institute.

FRANK FISHBOURNE BOYD (1875–1936)

Frank Fishbourne Boyd, a member of the A.S.M.E. since 1917, will be remembered most particularly as consulting engineer and general sales manager, from 1907 to 1917, of the firm of Robt. Wetherill & Co., Inc., Chester, Pa., engineers, iron founders, and builders of heavy power machinery, such as Corliss, marine, uniflow, blast-furnace and producer-gas engines, and pumping engines. The company also did heavy machine work in general, developed special machinery for industrial application, and manufactured high-pressure boilers and steel plate. In addition to his work in connection with these regular lines of the company, Mr. Boyd originated and managed the electric and plunger passenger elevator department.

Among Mr. Boyd's achievements were the design and installation of power plants for the Widener and Finance buildings, Philadelphia, Hotel Traymore, Atlantic City, N.J., Hershey Chocolate Company, Hershey, Pa., Lee Tire & Rubber Co., Conshohocken, Pa., Eastman Kodak Company, Rochester, N.Y., John A. Roebling & Sons, Trenton, N.J., and many other large companies in the East. He held professional engineering licenses in the States of New York, Florida, Arizona, and California, and left monuments of engineering skill to his memory in each of these states.

Shortly before his death Mr. Boyd retired as president of the Engineers Club of Los Angeles. At one time he was a member of the Manufacturers Club, of the Engineers Club of Philadelphia, and of the Riverton Yacht Club, Riverton, N.J., where he raced his own cruiser, the *Calvania*.

Mr. Boyd was born in San Francisco, Calif., on November 16, 1875, the son of Frank Alexander and Isabella (Fishbourne) Boyd. He was graduated from the Laurel Hill Military College, San Mateo, Calif., with a B.A. degree in 1893, and then took a special technical course in naval architecture at the Hoitt's Preparatory School in Burlingame, Calif., receiving a B.S. degree in 1894.

After a year of practical machine and electrical work at the Risdon Iron Works, San Francisco, he became a junior engineer on the S.S. *Zealandia*, of the Oceanic Steamship Company. Subsequently he served on the S.S. *St. Paul*, of the Alaska Commercial Company, and S.S. *China*, of the Pacific Mail Company, as assistant engineer.

Mr. Boyd interrupted his career in naval architecture and engineering design to serve in the Spanish-American War. He was assigned to duty in the U.S.A. Transport Service (Pacific Fleet), as chief electrical engineer on the transport ship *Grant*, and was a participant in the battle of Manila Bay. His ship later was sent to China in connection with the Boxer insurrection and other troubles in the Far East.

Leaving the army service in 1900, Mr. Boyd was associated until 1904 with J. M. Klein & Co., San Francisco, as consulting mechanical and electrical engineer and general superintendent of construction, for central-station and hydroelectric-power installations. During the next two years he was commercial engineer in the district office at San Francisco of the Westinghouse Electric & Manufacturing Co., engaged in the design and supervision of steam and hydroelectric power plants. From 1906 to 1908 he was power specialist and consulting engineer for the Allis-Chalmers Company, of Milwaukee, Wis., his duties taking him to numerous plants throughout the country.

Then came his long period with Robt. Wetherill & Co., Inc., terminated by the outbreak of hostilities between the United States and Germany in 1917. He reported for duty at League Island, Philadelphia, in April, 1917, and saw service throughout the World War. His first duty was to lay out the projected torpedo boat destroyer base at Lewes, Del., including the machine shops and shipways.

Following this, from the latter part of August, 1917, until early March, 1918, he was the Senior Engineer Officer of the U.S.S. *Jupiter*, then a collier, now the U.S.S. *Langley*, airplane carrier. The *Jupiter* was the first electrically propelled ship of the United States Navy. Mr. Boyd was aboard the *Jupiter*, carrying the first load of ammunition to France, when two torpedoes, fired at her by a German submarine, narrowly missed their mark.

Later Mr. Boyd was ordered to the U.S. Submarine Base at New London, Conn., to serve as engineer and repair officer, being the second officer in command at the Base, under Commander John Rodgers. His duty here from March 10, 1918, to March 1, 1919, was even more arduous, as he served frequently for seventy-two hours at a stretch, often finding, on being relieved for a rest, that some equally overworked and wornout submarine captain was asleep—boots and all—in his bed; and rather than disturb a brother officer, he would lay his own head on folded arms upon his desk. He received the sobriquet of "Old Iron Sides" for always being on duty.

His last duty in the Navy was as a member of the Naval Overseas Transportation Service, Ship Demobilization Board, returning to private owners the fleet of yachts and other vessels that had been used by the Navy Department for war service. He left the active service March 17, 1919, and was honorably discharged March 27, 1921, with the rank of Lieutenant-Commander (E), United States Naval Reserve Force.

After his war service, Mr. Boyd supervised the machinery installation in the thirty-seven 28,000 shp torpedo boat destroyers for the U.S. Navy built at the plant of the New York Shipbuilding Corporation, at Camden, N.J. He then took up the development of Diesel oil engines for marine purposes and was responsible for the layout and installation of machinery for the first commercial Diesel-electric motorship, the *Fordonian*, proving the feasibility of this class of drive for merchant vessels. He organized and directed the Marine Division of the Westinghouse Electric & Manufacturing Co. at New York, and laid out the Diesel-electric equipment of the Poughkeepsie and Highland Ferry Boat, an ice breaker. He built numerous Diesel-power and refrigerating plants in Florida, also large water-works plants in that state, employing mostly high-capacity direct-current turbine units with special automatic electrical control. During recent years he had specialized for the most part in industrial-engineering work, plant appraisals, and economic surveys.

His long record of intensely active work as an engineer continued to the day of his death, February 11, 1936, in Pasadena, Calif.

HENRY CHARLES BRIGGS (1883–1936)

Henry Charles Briggs was born on March 26, 1883, in Jersey City, N.J., son of Charles Henry and Sarah Augusta (Clark) Briggs. After completing his grammar-school education in Jersey City he took courses in steam- and marine-engineering subjects through the International Correspondence Schools, Scranton, Pa., graduating in 1905 as a marine engineer. He also held a United States marine license as chief engineer of ocean steamers of any gross tonnage and a New Jersey State license as chief engineer of power plants of any horsepower.

After a number of years in the merchant marine Mr. Briggs took the position of engineer on the steamer *Monmouth* of the Central Railroad of New Jersey in 1910. During part of 1912 and 1913 he was power engineer at Fort Wood, N.Y., then returned to the Central Railroad as chief engineer of the Communipaw (N.J.) power house. He continued in that position until the latter part of 1917, when he was engaged by the Texas Oil Company, New York, to conduct tests in power and various types of industrial plants to increase efficiency in operation through improved lubrication.

After completing that work early in 1918 he became assistant superintendent of maintenance and chief power engineer for the Westinghouse Electric & Manufacturing Co. at Lester, Pa., where he continued until the fall of 1919. He then returned to the employ of the Central Railroad of New Jersey as general mechanical foreman of the coal-dumping plant at Pier 18, Jersey City. He was in charge of maintenance and operation of two 125-ton McMyler coal dumpers.

He left the employ of the Central Railroad in the summer of 1927 to become supervising operator of the Holland Tunnels, New York. In June, 1932, he was made chief supervisor in charge of the maintenance of fans and rolling stock and operation of the tunnels. Late in November, 1935, he was promoted to assistant superintendent of the Operating Division, but the illness which led to his death on January 21, 1936, prevented his filling of that position.

Mr. Briggs became an associate member of the A.S.M.E. in 1926. He was also a member of the Junior Order of United American Mechanics, Hoboken, N.J., and a Mason. Married in 1911 to Clara Mabel Francis, he is survived by her and by two children, Mrs. Alfred J. Mendles of Wood-Ridge, N.J., and Edward W. Briggs, a student of aeronautical engineering at Auburn Polytechnic Institute.

CHARLES FREDERICK BRISTOL (1880–1934)

Charles Frederick Bristol, a member of the A.S.M.E. since 1916, was born at Madoc, Ontario, Canada, on March 24, 1880, the son of Wellington and Charity (Ketcheson) Bristol. He attended the Harbord St. Collegiate Institute, in Toronto, and was graduated from the Faculty of Applied Science of McGill University, Montreal, in 1908, with a B.Sc. degree in mechanical engineering.

Prior to entering McGill University Mr. Bristol served an apprenticeship at the Montreal plant of the American Locomotive Company, known as the Locomotive & Machine Co. of Montreal, from January, 1903, to September, 1904. He also worked there during the summers of 1905 and 1906. He spent the summer of 1907 as inspector of construction at the Moncton Shops of the Intercolonial Railway and during the year following his graduation was engineer in charge of the erection of the gas-engine power plant and other machinery at the Moncton Shops.

He then went to Vancouver, B.C., where he engaged in the design and construction of penstocks at the Stave Falls plant of the Western Canada Power Company for approximately two years. He was also engaged on the design and erection of their towers for the original high-tension transmission line from Ruskin. From 1911 to 1913 he was chief engineer and designer for the Segur Oil Refineries, Ltd., Vancouver, and also the British Columbia Fish Oil Refinery.

Returning to Montreal in 1913, Mr. Bristol entered the employ of Armstrong-Whitworth of Canada, Ltd., as assistant engineer to Matthew J. Butler, who had been helpful to him in connection with his engineering studies and at whose suggestion he later became a member of the A.S.M.E. Mr. Butler's memorial notice appears elsewhere in this issue. Mr. Bristol was put in general charge of the Mechanical Department, designing and supervising the erection of the boiler plant and electrical equipment, selecting machinery and tools, etc. Early in 1914 he was appointed general superintendent of the company, which sent him to England to make a special study of tool- and alloy-steel manufacture. This was the background for his later work as a metallurgist in the manufacture of steel by electric and crucible processes.

Late in 1919 Mr. Bristol again went to Vancouver, where he engaged in private consulting work and later became mechanical engineer and metallurgist for the Vancouver Engineering Works.

After leaving that company in July, 1926, he was engaged by the National Supply Company of California to improve output and reduce costs at the Union Tool Division, Torrance, Calif.

Since 1931 he had been connected with the Latrobe Electric Steel Company, Latrobe, Pa., manufacturers of electric-furnace steels, as a metallurgist. He died at Latrobe of a heart attack on April 23, 1934. His widow, Jessie D. (Ross) Bristol, whom he married in 1922, survives him.

OLIVER CROMWELL BROOKS (1876–1936)

Oliver Cromwell Brooks, who was born in Philadelphia, Pa., on June 15, 1876, came of engineering families on both his mother's and his father's sides. His mother, Julia Carrie Bormann, was the daughter of Johann Carl Eduard Bormann, a graduate of the Royal Technical Institute of Dresden in 1837. His father, Arthur Gilman Brooks, and his uncle, Emery James Brooks, were both engineers in the United States Navy during Civil War times. Three of his cousins, William H. Sayre, Allen Brooks Cuthbert, and James Emery Brooks, were mechanical or civil engineers.

Oliver Brooks attended the Philadelphia public schools, including the Technical Manual Training School, and also had private instruction. After a few months with the Keen Sutterlee Company, Philadelphia, he was engaged by the Camden (N.J.) Lighting & Heating Co., to work on plans for new lighting and power plants. Later he was put in charge of erection, laying out of all work, and installation of boilers, engines, generators, etc. He designed special tubular boilers and a self-sustaining steel stack. During the period spent with this company he also mapped lighting and power circuits in Camden, made a survey for water lines, wells, etc., for the town of Merchantville, N.J., and worked on glass-furnace designs and special molds used in the manufacture of bottles.

In May, 1898, Mr. Brooks entered the portland cement industry as draftsman for the Alpha Portland Cement Company, Easton, Pa. After a few months he was made assistant superintendent and during the latter half of the nearly twelve years he was connected with the company, he served as chief mechanical engineer, in full charge of the design and erection of all work at their several plants.

The next company with which Mr. Brooks was connected was the Texas Portland Cement Company of Dallas. He became assistant superintendent there in May, 1910, and continued in that position until early in 1912. He left to enter the consulting field in Dallas in

association with L. L. Griffiths, who had been general superintendent of the Texas Portland Cement Company. Mr. Griffiths was president and Mr. Brooks secretary of the L. L. Griffiths Engineering Co. The announcement of the formation of the company brought Mr. Brooks many letters, from one of which may be quoted the following comment on his previous work:

"There is no question of your ability to successfully engage in the work contemplated, starting as you did as one of the pioneers in the portland cement industry in the United States. Your twelve years' connection with the Alpha Portland Cement Company as chief engineer, during which time you had charge of the design and building of new plants, altering and improving old ones, and selecting and installing the machinery used in the mills and power houses, has given you a broad and varied experience in civil, mechanical, and electrical engineering, which will make for the success of the work you undertake, and for the clients retaining you."

The company was of short duration, however, as better opportunities soon developed for both men. Mr. Brooks went to New York as engineer for the Robins Conveying Belt Company. After several years there he went to Baltimore, Md., where he was connected first with the Baltimore & Ohio R.R. and then with the International Conveyor Corporation, for which he was chief engineer. Later his work for this company took him back to New York.

In 1922 Mr. Brooks again became identified with the portland cement industry as assistant superintendent and chief mechanical engineer of the Michigan State Cement Plant at Chelsea, Mich. Since he left there in 1927 he had not worked continuously. He was consulting engineer for the Amtorg Trading Company in New York for a time and more recently had worked for the firm of Hickman and Williams, of St. Louis, Mo. He died in New York on April 29, 1936.

Mr. Brooks had been a member of the A.S.M.E. since 1912 and was a Mason. His recreations were reading, fishing, and hunting. He had never married.

THOMAS VAN WYCK BRUSH (1885–1935)

Thomas Van Wyck Brush, superintendent and chief engineer of the Palmer House, Chicago, Ill., died in that city on April 1, 1935. He was born in Brooklyn, N.Y., on January 6, 1885, son of Thomas Van Wyck and Emma E. Brush. He attended the Brooklyn Public Schools and took courses through the International Correspondence Schools, Scranton, Pa. His early engineering work was in New York, first as electrical engineer in charge of plant maintenance in the Singer Building from 1904 to 1910, and then as engineer in charge of plant operation and maintenance and advisory engineer on matters relating to mechanical and electrical construction of the Lord & Taylor department store at Fifth Avenue and Thirty-Eighth Street.

In 1914 Mr. Brush went to Trenton, N.J., to enter the employ of the Globe Rubber Tire Manufacturing Company. He was there for five years, serving as engineer in charge of maintenance and operation of building, power-plant, and mill machinery, and laying out and equipping two buildings for tire manufacturing and also a machine shop. He returned to New York to take the position of chief engineer at the Plaza Hotel, and remained there until he went to the Palmer House in 1924.

Mr. Brush became an associate-member of the A.S.M.E. in 1921. He was also a member of the National Association of Stationary Engineers, New York Building Superintendents Association, and the Masonic fraternity, being a member of the Scottish Rite and Shriners. He had served in the New York State Militia.

Mr. Brush married Nora F. Piercy in 1906 and is survived by her and by two children, Claire E. and Victor V. Brush.

JOHN H. BUCKLEY (1872–1935)

John H. Buckley, who had been associated with the Otis Elevator Company, New York, N.Y., as a designer and mechanical engineer, the greater part of the time since 1902, died on March 3, 1935, at the United States Veterans Hospital, the Bronx.

Major Buckley was born at Wassaic, N.Y., on November 21, 1872. He supplemented his public schooling with correspondence courses and private tuition and qualified as a master operating engineer. His early experience was with the Brown & Sharpe Manufacturing Co., Providence, R.I., U.S. Rubber Co., Trenton Oil Cloth & Linoleum Co., as master mechanic, Bridgeport Brass Company, in charge of a rod and tube mill, and Hodgman Rubber Company, as chief engineer.

He was with the Otis Elevator Company from 1902 until the beginning of the World War. He served as a major in the Ordnance Department through the war and from 1930 until his death was Commanding Officer of the 40th Engineers, U.S.A. From 1922 to 1929 he was inspector of construction for the Interborough Rapid Transit Company, and since then had been with the Otis company.

Major Buckley became a member of the A.S.M.E. in 1912. He was a former commander of Yonkers Commandery, Knights Templar, and a Royal Arch Mason. Two sons, Chester and Will Buckley, survive him.

LOUIS HENRY BURKHART (1869–1935)

Louis Henry Burkhart, chief engineer of the Struthers Wells-Titusville Corporation, Warren, Pa., died at the St. Francis Hospital, Pittsburgh, Pa., on March 5, 1935. He was born on May 23, 1869, in Louisville, Ky., a son of John and Emma (Johnson) Burkhart. At the age of thirteen he moved to Norborne, Mo., where he completed his preparatory education. He took the normal course at the University of Missouri, after which he taught in the Missouri schools for several years. He then returned to the University to continue his studies, graduating in 1897 with a B.S. degree in mechanical engineering.

Immediately following his graduation Mr. Burkhart accepted a position with the Struthers-Wells Company. For a short time he had charge of the Estimating and Order Departments, but soon became engineer in charge of the Plate Department. He left the company in March, 1903, to become superintendent of the Crescent Iron Works, Springfield, Mo., but returned in August of that year as chief engineer of the company. He had served as a director of the Struthers Wells-Titusville Corporation since February, 1926.

During his many years with the Struthers-Wells Company, Mr. Burkhart designed and developed a great deal of equipment and apparatus. A list which he prepared at the time he became a member of the A.S.M.E. in 1921 included the pressed-metal transformer cases on which he secured a patent in 1912 and which were adopted by the large electric companies; wood-distillation apparatus, particularly that for the Seaman process for the utilization of wood waste; special activating apparatus for the production of carbon for gas masks; and equipment for explosives factories, sugar centrals, turpentine plants, soap factories, carbon-black plants, and by-product coke plants. He had also developed other special apparatus for the chemical and oil industries, glass-making machinery, hydraulic equipment, such as penstocks, flumes, and distributors, self-supporting stacks, elevated tanks and towers and water towers, fire- and water-tube and special waste-heat boilers, steam and gas engines, and dies and formers for pressing, flanging, and drawing, and had directed the manufacture of structural steel for factories, etc., and selected machinery and equipment for manufacturing plants. This list, to which later years added many new items, indicates the breadth of his experience and his value to the Struthers-Wells Company.

Mr. Burkhart was a member of the American Welding Society and of its Conference Committee to the Subcommittee on Welding, of the A.S.M.E. Boiler Code Committee. He had also served as a member at large of the Joint Research Committee on Welding of Pressure Vessels, which was discharged in 1931, and of its Subcommittee on Procedure.

He was a pioneer in the development and application of gas and electric welding to steel-plate construction and special machinery, as well as the application and use of the modern metals and alloys in welded construction.

Mr. Burkhart was also active in community affairs, a deacon in the First Baptist Church, director of the Warren Y.M.C.A., and also a director of the Warren National Bank for several years. He had served as a member of the school board during the time that a reconstruction program was carried out which resulted in the community's having some of the finest public schools in the section. For many years he was active in the Kiwanis Club.

Mr. Burkhart married Elizabeth Hoddle, of Norborne, Mo., in 1898, who survives with two sons, William H., of Baltimore, Md., and Louis H., Jr., of Union City; two grandchildren, Diane and Cynthia Burkhart; and two brothers, Richard W., of Tulsa, Okla., and John Burkhart, of Kansas City, Mo.

BRUCE ARTHUR BUTCHER (1900–1934)

Bruce Arthur Butcher, who became a junior member of the A.S.M.E. in 1928, died on August 6, 1934, at Prescott, Ariz., after being in poor health for some time. He is survived by his widow, Hazel Martha (Raver) Butcher, whom he married in 1929, and by their son, Allen Arthur Butcher.

Mr. Butcher was born at Jackson, Mich., on August 6, 1900, son of Arthur Lewis and Jessie (Anderson) Butcher. After graduating from the Jackson High School he entered the University of Michigan, from which he received a B.S. degree in mechanical engineering in 1923. He was elected to membership in the honorary fraternities,

Sigma Xi and Tau Beta Pi. He was employed during the following year as a draftsman, first by the Reo Motor Car Company, Lansing, Mich., and later by the Hayes Wheel Company, in Jackson, for which he did layout work on woodworking machinery.

In May, 1924, he began work in the Mechanical Engineering Department of the Commonwealth Power Company, Jackson, collecting, computing, and recording operating data for plants and their equipment. Ill health prevented him from working part of 1926 and 1927, and subsequently he was engaged in detail and layout work on outboard motors for the Lockwood Division of the Outboard Motors Corporation, Jackson. In August, 1928, he returned to the Commonwealth Power Company to work on layout of piping for power plants. After completing his work for that company the following year he was not regularly employed, but worked at different times for the Engineering Department of Allied Engineers, Inc., Jackson, on power plant piping layouts; for the Republic Iron & Steel Co., Youngstown, Ohio, in electrification and waste-heat investigation; for the International Nickel Company, at Copper Cliff, Ontario, Canada, in an investigation of a waste-heat installation; and as draftsman for Stevens & Wood, Inc., Jackson, and the Firestone Tire & Rubber Co., Akron.

MATTHEW JOSEPH BUTLER (1856–1933)

Matthew Joseph Butler was born of Irish parentage, at Deseronto, Ont., Canada, on November 19, 1856, the son of Tobias and Elizabeth (McVey) Butler.

After completing the public school course, he entered University College, Toronto, Ont., Canada. In 1874, on leaving University, he was articled to the engineering firm of Evans and Bolger, at Belleville, Ont. On the completion of his articles in 1878, Mr. Butler entered private practice and, in 1880, had charge of the extension of the Bay of Quinto Railway. In 1882, he was engineer in charge of the location and construction of the Thousand Islands Railway, and during the following five years he was active in railway location and construction projects throughout Ontario.

In 1887, Mr. Butler entered the service of the Atchison, Topeka and Santa Fé Railway Company in Kansas and Colorado. In 1890 he returned to Canada to become manager and part owner of the wood-pulp mill, at Rivière du Lac, Que., Canada, and during the following five years was identified with a number of railway and industrial projects, including a cement factory, lumber mills, and brick plants. At the same time he held the position of chief engineer for the Bay of Quinto Railway during the construction of the Sydenham Branch of that system. In 1895 he built the Oshawa, Ont., Electric Railway.

From 1895 to 1900, Mr. Butler was engaged in general consulting work and, during this period, he devoted himself to the study of law. He was graduated from Kent College of Law, Chicago, Ill., with the degree of doctor of laws, and was admitted to the Bar of the State of Illinois in 1897.

From 1900 to 1904, he was in charge of the building of the Hillsborough Bridge at Charlottetown, Prince Edward Island, during which difficult conditions were encountered, necessitating the sinking of pneumatic caissons 100 ft below tide level. The successful accomplishment of this project was always looked upon by Mr. Butler as his most noteworthy professional work. It brought him to the notice of the Canadian Government by which he was engaged as assistant chief engineer of the 1800-mile section of the Grand Trunk Pacific Railway from Moncton, N.B., to Winnipeg, Man., Canada. In addition to the regular duties which this position demanded, Mr. Butler acted as adviser to the Grand Trunk Pacific Railway Company in drawing up field specifications under which the road was constructed. This position was held from September, 1904, until May, 1905, when Mr. Butler resigned to receive the appointment of deputy minister and chief engineer of the Federal Department of Railways and Canals of Canada, and also chairman of the Board of Management of the Canadian Government Railways. In January, 1910, he resigned these positions to become second vice-president and general manager of the Dominion Iron and Steel Company, Dominion Coal Company, and Associated Companies. This position he filled until 1913. During his tenure of office as general manager, an extensive construction campaign, including important additions to the steel works and mines, was undertaken and completed.

On leaving the Dominion Steel and Coal Companies he became associated with the Sir W. G. Armstrong-Whitworth Company and was appointed chief engineer and superintendent of construction for its Canadian plant, which was begun at Longueuil, Que., in 1913 and completed in July, 1914, just before the outbreak of the World War. Mr. Butler then became managing director and designed and built additions to the plant for the manufacture of high-speed crucible steel and installed electric smelting furnaces to supply many of the special steels required during the war period.

On account of ill health, he resigned from this position in 1918, and to the time of his death on June 22, 1933, was practically retired from all professional work, except to act as adviser in special lines of construction work.

In 1880, Mr. Butler was married to Lorette M. J. Shibley, who predeceased him. He is survived by two daughters.

Notwithstanding his very busy and varied career, Mr. Butler yet found time, in 1900, to act on the Royal Commission on Municipal Assessment and Taxation, and, in 1907, to become a member of the Royal Commission of Forestry. He was a Fellow of the Royal Society of Arts; a member of the Royal Art Society of Canada; and a member of the Institution of Civil Engineers, of London, England, the Engineering Institute of Canada (president in 1914), the American Society of Civil Engineers, and The American Society of Mechanical Engineers, which he joined in 1912. He was decorated by His Majesty King Edward VII as Companion of the Order of St. Michael and St. George for distinguished service in Canada. In 1917, he received the degree of doctor of laws from St. Francis Xavier College, Antigonish, N.S., Canada.

Mr. Butler's varied professional experience, his wide range of technical reading, and his keen interest in engineering problems, made his advice on all professional subjects particularly valuable. He showed great interest in young members of his own profession and was always glad to assist and encourage them in their work. The sympathy and understanding he showed in such cases endeared him to all engineers who were fortunate enough to be associated with him. He was always an interesting and pleasant companion, a faithful friend, and a true gentleman. His death removes one of the last of the old school of engineers who played so important a part in the railway and industrial expansion of Canada. [Memoir prepared by D. H. McDougall, Esq., Stellarton, N.S., Canada, for the Transactions of the American Society of Civil Engineers.]

LOUIS GEORGE CARPENTER (1861–1935)

Louis George Carpenter, formerly state engineer of Colorado and internationally known as an authority on irrigation, of which he was one of the early and chief developers in the Rocky Mountain States, died at St. Luke's Hospital in Denver on September 12, 1935, after a long illness following a stroke of paralysis at his home in that city.

For nearly half a century he fought Colorado's battles for her rightful share from interstate streams. His contributions to the increasing knowledge of irrigation brought him high honors from France, Canada, and the Government of the United States. His books on irrigation are standard reference works used in universities in many lands.

Born on March 28, 1861, at Orion, Mich., he came from a family that pioneered in the development of America. His father, Charles Ketcham Carpenter, was descended from three of the original proprietors of Providence, R.I. His mother, Jeannette (Coryell) Carpenter, traced her descent from the earliest Dutch settlers of the seventeenth century in New Netherland, and from a French Huguenot ancestor who came to America in 1700.

Louis Carpenter attended Michigan Agricultural College, where he obtained his B.S. in 1879 and his M.S. degree in 1883. His fraternity was Delta Tau Delta. He did postgraduate work at Michigan University in 1884 and at Johns Hopkins University from 1885 to 1888. He was instructor in mathematics at Michigan Agricultural College, 1881 to 1885, and then until 1888, assistant professor of mathematics.

In February, 1887, at Jackson, Mich., he married Mary J. C. Merrill of Lansing, and the following year took his young wife to Colorado, having accepted the appointment of professor of engineering and physics at the Colorado Agricultural College at Fort Collins, a position he held from 1888 to 1911.

During this period his reputation as an irrigation engineer circled the globe and brought him honors and medals from foreign governments. His first year at Fort Collins saw him organizing the first systematic instruction in irrigation engineering and investigation in that line in the United States, and in addition to becoming irrigation engineer of the Colorado Experiment Station, he was the irrigation expert for the United States Department of Agriculture. He was director of the Experiment Station from 1899 to 1910 and wrote numerous administrative and technical reports, papers on investigations, engineering, irrigation, etc., that became authoritative works on these subjects.

France bestowed upon him the ribbon and order of Chevalier du Mérite Agricole in 1895, and in 1900 at the Paris Exposition he was awarded a gold medal. Another gold medal was bestowed on him in recognition of his contributions to the science of irrigation, at the Portland Exposition.

In addition to his duties as a member of the faculty of the State

Agricultural College, he did much important work for the United States Government. He was special agent for the Department of Agriculture, making the arduous artesian well investigation in 1890, and in 1900 wrote the voluminous "Reports on Artesian Wells of Colorado and New Mexico," the investigation report for Congress, and predicted the limits of the artesian basin of the San Luis Valley, Colorado.

He was sought by British Columbia to draw its Provincial irrigation code. In 1907 and 1908 he was a member of the irrigation commission of that Canadian province to determine the foundations for a new water code, which was adopted by the Parliament. An already internationally recognized authority in irrigation litigation, he was the principal expert in the celebrated international case of the United States versus the Rio Grande Dam, this concerning the great Elephant Butte project.

He was the consulting engineer and irrigation expert for the State of Colorado, 1903 to 1907, in interstate litigation, appearing in the suits brought by Kansas and by Wyoming against Colorado. In the Kansas suit he made the physical and engineering investigation to present to the United States Supreme Court, and was the consulting engineer preparing the case for Colorado in the suit brought by Wyoming. In 1920 both sides in the dispute over the waters of the North Platte in the case of the United States versus Wyoming, selected Dr. Carpenter as arbitrator and he refereed the claims, counterclaims, and water questions between Wyoming and the United States. He was selected by both parties as referee and chairman of the board of arbitration to settle the electric lighting controversy in 1907 between Colorado Springs and the Colorado Springs Lighting Company involving determination of disputed engineering questions of contract, making the leading case.

Pueblo, Canon City, and Salida engaged him as their expert in stopping the pollution of the Arkansas River through mining operations. He was consulting engineer for many important dams, irrigation, and hydraulic enterprises, and was a member of the commission to locate the Gunnison Tunnel.

In a life extraordinarily busy, he found time to write authoritative volumes that are recognized the world over, among them "Water Measurement," introducing the trapezoidal weir, now universally in use; and "The Duty of Water," which embodies the report of the first systematic investigation. His comprehensive work on seepage, measurement of losses, etc., from reservoirs, canals, and streams embraced findings covering many thousand of miles. He supplemented his valuable volume on "Forests and Snow" by extensive winter observations. His "Meteorological Records" was of extensive character, including highly important new facts on solar radiation. These contributions led to his appointment as coeditor of the Standard Dictionary.

Dr. Carpenter was state engineer of Colorado, 1903 to 1905, his office having charge of the entire irrigation system. The late Enos Mills, of Estes Park, was employed by Dr. Carpenter as a snow observer for several years and his winter journeys through the forests and above the timberline gave Mills the material that afterward formed the magazine articles and books bringing him fame as a naturalist. Dr. Carpenter was struck with the occasional poetic and vividly descriptive phrases occurring in the otherwise dry reports and, himself a writer, he encouraged the young observer to tell the world of the wild life he encountered in the high mountains.

Dr. Carpenter served his state and country in the World War as a member, one of the first seven, of the Colorado State Council of Defense, the first of such boards organized in the United States. During the two years, 1917–1919, he was director of the department of publicity and information of the Council, and was also vice-chairman of the Explosives Board.

Many honors came to him. He was president of the American Society of Irrigation Engineers, 1893–1895; vice-president, American Association of Agricultural Colleges and Experiment Stations, 1900; chairman of the engineering group of the Denver Chamber of Commerce, 1913–1926; president, Colorado State Historical Society, 1918–1920; president, Colorado Scientific Society, 1925–1928; president, Colorado Engineering Council, 1923; Fellow of the American Association for the Advancement of Science; delegate to the World's Engineering Congress, Tokio, Japan, 1929. He was a member of the American Institute of Electrical Engineers, the Society for the Promotion of Engineering Education, one of the incorporators and an honorary member of the Michigan Engineering Society, member of the American Society for Testing Materials, The American Society of Mechanical Engineers (since 1923), Colorado Academy of Sciences, Society of the American Revolution, Society for the Promotion of Agricultural Science, and the Forestry Association.

From 1908 to 1914 he was a director of the First National Bank of Fort Collins, and from 1910 to 1915, president of the Fort Collins Building and Loan Association. Actively aiding in the development of the rich agricultural districts in northern Colorado, Dr. Carpenter was president of the Windsor Land Company and the Tongue Creek Orchard Company.

He was an authority on the history of Colorado, and his library, one of the largest private collections in the West, contains many rare first editions and books of the early West that are eagerly sought by collectors. Recently he presented the Denver Public Library with hundreds of technical and scientific works of great value. As he was versed in French and Italian and other languages, his library contains the leading works in first printings of Europe's noted scientists.

For years he had devoted his spare time to translating Leonardo Da Vinci and had assembled a notable collection of the works of that remarkable man of many brilliant gifts. His fame as an authority on Da Vinci brought him correspondence from literally over the world. So well was he known to French scholars that Marshal Foch, French World War hero, when in Denver in 1921, sought Dr. Carpenter out for a visit.

Mrs. Carpenter died on July 4, 1921. There were two children by this marriage. Charles L. Carpenter, born in 1889, and recently deceased, was captain of a motor truck company in the A.E.F. in France, and participated in American major engagements during the War. A daughter, Jeannette C. (Mrs. Roe) Emery, survives him, as does also his second wife, Katherine M. (Warren) Carpenter, three sisters, a brother, Judge W. L. Carpenter, Detroit, Mich, and five grandchildren. Another brother, who died in 1919, was Professor Rollo C. Carpenter, a former vice-president of the A.S.M.E. and long a prominent member of the faculty of Sibley College, Cornell University. The two brothers were associated in consulting practice for a number of years.—[Memorial prepared by THOMAS B. STEARNS and FRANK E. SHEPARD, Denver, Colo., and L. D. CRAIN, Fort Collins, Colo., Members of the A.S.M.E.]

JOHN RICHARD CAVE (1879–1935)

John Richard Cave, for more than twenty years fuel engineer for the Board of Education of the City of New York, died at his home in Bayside, Long Island, N.Y., on August 18, 1935.

A native of England, Mr. Cave was born at Liverpool on June 13, 1879, son of John Richard and Elizabeth Taylerson (Barnes) Cave. He was educated in England, securing his preparatory schooling at Truro College and studying civil engineering at Trinity College, Dublin, and mining and mechanical engineering at the University of Cornwall.

Mr. Cave's first position was that of electrical and mechanical engineer for the French telegraph and cable bureau. He began work for the City of New York as draftsman for the Topographical Bureau in the Borough of Queens and after a short time there became assistant engineer in the Department of Water Supply, Gas and Electricity. He was in charge of engine and boiler construction, inspection, and erection, but was chiefly engaged in investigating conditions relative to steam-plant economy. He conducted fuel tests at the Department's testing station at Flushing, N.Y., and at the Ridgewood Pumping Station in Brooklyn. In 1912 he was made engineer-in-charge of the Fuel Economy Bureau of the Department.

Appointed fuel engineer to the Board of Education in 1913, Mr. Cave devoted the remainder of his life to problems relative to the economic consumption of fuel in the boiler plants of the city's school system. He made fuel-waste investigations, and recommended types of fuel, equipment, and installations best suited to the growing needs of the system.

At the time of his death he was consulting fuel engineer to the Board of Estimate and Apportionment and fuel engineer of the Board of Education. He served as a Deputy Sheriff of Queens County from 1932 to 1935.

Mr. Cave invented an automatic stoker for domestic heating installations and a driving mechanism, similar to free-wheeling, for motor vehicles.

Keenly interested in yachting, Mr. Cave had been a member of the Bayside Yacht Club since 1912 and had served on several of its committees. He became a member of the A.S.M.E. in 1913.

Surviving Mr. Cave are his widow, the former Miss Wilhelmina Marie Gerken, whom he married in 1901, and three children, John Richard Cave, Jr., and two daughters, Vivian Dorothy and Marjory Willa Cave.

JOHN WELKER CHINN (1903–1936)

John Welker Chinn, a member of the technical staff of the Bell Telephone Laboratories, New York, N.Y., engaged in the development of microphones and electrical vibration instruments, died on February 7, 1936, following an operation for appendicitis.

Mr. Chinn was born at Odon, Ind., on August 15, 1903, a son of Al-

fred Price and Eva G. Chinn. He prepared for college at the Garfield High School, in Terre Haute, Ind., and was graduated from Rose Polytechnic Institute with a B.S. degree in 1930. Immediately following his graduation he joined the Systems Development Department of the Bell Telephone Laboratories. Two months later he was transferred to the Transmission Instruments Department of the Research Department where for two years he was connected with the development and improvement of electromechanical recorders for sound on disks and with the development of apparatus for blind flying. Since 1932 he had been with the transmitter groups and had worked on development problems relating to the handset transmitter and on analytical design studies of transmitters, receivers, and handsets.

Mr. Chinn became a member of the Student Branch of the Society while attending Rose Polytechnic Institute and a junior member of the Society in 1930. He also belonged to the Theta Xi fraternity. He was a first lieutenant in the 439th Engineers Officers Reserve Corps. He is survived by his widow, Mrs. Jessica P. (Taylor) Chinn; a son, John Sidney Chinn; his mother; and two sisters and two brothers.

MARTIN H. CHRISTOPHERSON (1866–1935)

Martin H. Christopherson, safety engineer and director of service for the New York State Insurance Fund, died at his home in Yonkers, N.Y., on September 9, 1935. He was connected with the elevator industry in numerous capacities from the time he entered the Crane Elevator Company as an apprentice until his death, a period of 55 years of continuous service.

Mr. Christopherson had served the New York State Labor Department since 1917, when he was appointed a member of the State Industrial Council. In 1921 he was made deputy industrial commissioner, a post which he held until he became connected with the State Insurance Fund in 1923. He also was a member of The General Supervisory Committee having charge of the Annual State-Wide Accident Prevention Campaigns of the Associated Industries of New York State, Inc.

Born at Horten, Norway, on June 6, 1866, he was brought to the United States by his parents when he was three years old. He attended high school in Chicago and began work with the Crane Elevator Company in that city. He was manager in charge of manufacturing for the Otis Elevator Company, at the Chicago works, from 1905 to 1909, and during the next nine years served the company as general works manager in charge of all works and engineering, with headquarters at the New York office.

During the years 1918–1920 Mr. Christopherson was manager of the Otis Ordnance Works at Chicago, supervising the manufacture of 240-mm French howitzer recuperators for the U.S. Ordnance Department. In 1920 he became president of the Davenport Manufacturing Company, Davenport, Iowa, and equipped the plant for the manufacture of internal-combustion oil engines. He served the Otis Elevator Company in the capacity of consulting engineer from 1920 to 1934.

Mr. Christopherson became a member of the A.S.M.E. in 1923 and rendered valuable service on a number of its committees. His experience in the manufacture of elevators made him a very acceptable chairman of the Special Research Committee on Elevators, and he was also a member of the Nominating Committee of that group.

He represented the International Association of Industrial Accident Boards and Commissions on the Sectional Committee on Safety Code for Elevators, and was chairman of its Subcommittee on Research, Recommendations, and Interpretations. He was a member of the original committee known as the A.S.M.E. Committee on Protection of Industrial Workers, which developed the first edition of the safety code for elevators in 1921.

Mr. Christopherson also represented the International Association of Industrial Accident Boards and Commissions on the Sectional Committee on a Safety Code for Conveyors and Conveying Machinery and served on its Subcommittee No. 3 on Gravity Conveyors and Chutes, Live Roll Conveyors, and Subcommittee No. 4 on Spiral and Track or Scraper Conveyors.

For four years Mr. Christopherson served on the A.S.M.E. Committee on Safety. He was a member of Subcommittee No. 4 on the Use of the ASA Code Versus State Codes, of the Sectional Committee on a Safety Code for Mechanical Power Transmission Apparatus, and represented the A.S.M.E. on the ASA Safety Code Correlating Committee, taking a leading part in its activities. He was also a member of the National Safety Council.

Surviving Mr. Christopherson are his widow, Ida G. (Hanson) Christopherson, and five children, Mrs. B. D. Cannon, of Milford, Conn., and Dorothy, Marvin T., Robert, and Harold, all of whom have taken the name of Christy.

JOHN WILLS CLOUD (1851–1936)

John Wills Cloud, who became a member of the A.S.M.E. in 1880 and who was formerly chairman of the Westinghouse Brake Company, Limited, London, England, died on January 14, 1936. He had been connected with the company for 46 years, from the days when George Westinghouse and H. H. Westinghouse were actively engaged in developing the Westinghouse brake.

Mr. Cloud was born at Woodbury, N.J., on October 27, 1851. He was graduated from the Lawrence Scientific School of Harvard University in 1873, with a B.S. degree. He studied both civil and mechanical engineering, but devoted his final year exclusively to the latter.

In 1874 Mr. Cloud entered the employ of the Pennsylvania Railroad in the Motive Power Department at Altoona, Pa. The following year he was placed in charge of the testing of materials used in construction. He also investigated the efficiency of engines and fuels, examined railway supplies in general, and tested patented devices applicable to railways. In 1879 he was given the title of engineer of tests, and continued in that position until 1887. During the next two years he was superintendent of motive power for the Erie Railroad, Buffalo, N.Y.

Mr. Cloud's association with the Westinghouse companies began in 1889 as western representative, with headquarters in Chicago, and he continued in that capacity for ten years. In 1899 he went to London to become vice-president of the Westinghouse Brake Company, Limited, an office which he held until 1920. From then until 1931 he was chairman of the company, and from 1931, after his retirement from the chairmanship, he had been an active director of the London company. He made outstanding contributions to air-brake progress and was the inventor of the quick-acting triple valve for European service. He traveled widely and took a keen interest in the development of the Westinghouse brake on the Continent.

In addition to his membership of more than fifty-five years in the A.S.M.E., Mr. Cloud was a Fellow of the American Association for the Advancement of Science and a member of the Institution of Mechanical Engineers, the Royal Society of Arts, The Franklin Institute, and other scientific and engineering societies. He served as secretary of the Master Car Builders' Association when located in Chicago. Since taking up his residence in London he had become a British citizen.

FRANK J. COADY (1893–1934)

Frank J. Coady, who was killed in an automobile accident on September 29, 1934, was born in Philadelphia, Pa., on June 8, 1893. His parents were Harry Stephen and Anna C. Coady. Supplementing his four years at the Roman Catholic High School in Philadelphia, he took evening courses at the Central High School for a year, studied electrical engineering at Lehigh University for four months, and took the mechanical engineering course at Drexel Institute.

Mr. Coady's first position was with the Midvale Steel Company, Philadelphia, from 1912 to 1916, his duties including estimating for machine work, inspection of patterns and castings, general iron and steel foundry work, and supervision of cupola and annealing furnaces. During the next four years he was engaged in shipbuilding, chiefly as draftsman on engine and boiler-room layouts. He worked successively at the Navy Yards at Norfolk, Va., and Philadelphia, the Chester Shipbuilding Company, Bethlehem Shipbuilding Company, New York Shipbuilding Company, Camden, N.J., and Federal Shipbuilding Company, Kearney, N.J.

In 1920–1921 he was designing draftsman, in charge of engine and boiler-room piping and machinery layouts, at the Vulcan Iron Works, Jersey City, N.J., and from then until 1923 was draftsman on aerial tramways for the American Steel & Wire Co., Trenton, N.J. During the remainder of his life he was connected with the National Airoil Burner Company, Philadelphia, for the first three or four years as designing draftsman for various types of oil-fired furnaces and thereafter as chief engineer in charge of designing oil burners and making oil-burning layouts for power plants, the steel, glass, and ceramic industries, iron and brass foundries, refineries, etc.

Mr. Coady became an associate member of the A.S.M.E. in 1925 and belonged to the Engineers Club of Philadelphia. He is survived by his widow, the former Miss Josephine C. Dickel, whom he married in 1925.

THOMAS FRANCIS COLLINS, JR. (1911–1935)

Thomas Francis Collins, Jr., was born at Astoria, N.Y., on March 11, 1911. His mother's maiden name was Louise Catherine Eberle. He prepared for college at the William H. Hall High School in West

Hartford, Conn., and was graduated from the University of Vermont in 1933, with a B.S. degree in mechanical engineering. He was prominent in athletics at the University, playing football, basketball, and baseball. He was elected to membership in Sigma Nu fraternity and in the Gold Key and the Key and Serpent honor societies; was the first president of the "V" Club, permanent president of his class, and college senator; and was voted the most popular member of his class.

Following his graduation he entered the employ of the Hartford Machine Screw Company, engaging first in time-study work and later becoming foreman of the special order department, the position which he held at the time of his death on May 14, 1935.

Mr. Collins became a junior member of the A.S.M.E. in 1933.

FREDERICK NEVIUS CONNET (1867–1935)

On June 18, 1935, there passed away in Providence, R.I., at the age of sixty-seven, a member of the A.S.M.E. whose modesty of bearing was out of all proportion to his accomplishments as an inventor and an engineer. Frederick Nevius Connet left behind him enduring contributions to practical mechanics and hydraulics in a large number of devices of direct service to mankind. To the many who sought his advice on a great variety of questions he gave freely, but never without first keenly analyzing the problem, always illustrating graphically the conditions and the various steps in the solution. He often remarked he "could not talk without a pencil" and his unusual ability for sketching mechanical assemblies aided materially in clarifying these discussions.

Mr. Connet was born on October 16, 1867, at Flemington, N.J., the son of Andrew J. and Joanna (Nevius) Connet. He received his early education in the public schools at Flemington. Later he attended Reading Academy, and in 1889 was graduated from Stevens Institute with the degree of mechanical engineer. His excellent thesis upon the then new subject of compounding locomotives predicted unusual talents. He immediately entered the employ of Builders Iron Foundry, Providence, where, with the exception of one year at the Providence Engineering Works, he carried out his life's work.

Then, as now, the company turned out a large variety of special, often intricate, work, requiring excellence of design plus precision in casting and machining. The young engineer thus was met at once with a broad challenge; how completely he made good is still amply evident in the old files of drawings and in the recollections of the few still living with whom he was associated during this time. Space limitation prevents a detailed recital; it is adequate to say that his ingenuity contributed most importantly to details of design and shop-production methods of such material as 12-in. breech-loading rifled mortars, 123 of which were made in 1891 for the United States seacoast defense; the Rider-Ericsson hot-air engine; the Lowry cotton-bailing press; 13 barbette gun carriages for the U. S. Army; and the well-known Rice and Sargent steam engines.

In 1901 an order was taken from a New London shipbuilding concern for a special steel-plate planing and scarfing machine for the edges of boat plates. Previously it had been the practice to chisel-calk laboriously the edges of the overlapping plates at the joints to make them tight. This machine was designed by Mr. Connet to so face the edges of these long plates that they could be hammered down.

All the details of design of a complex screw-propeller planer for the U.S. Navy were worked out by Mr. Connet in 1902 and the machine was built in the Providence shops. The approximate dimensions were 9 ft high × 5 ft × 8 ft; the weight 10 tons. It operated on rough bronze propeller castings of two, three, or four blades with diameters varying from 5 in. to 24 in. It finished both the front and backs of the blades to the required pitch and form. The Navy was thus enabled to carry out many experiments of great importance upon moving ship models 20 ft long in the 500-ft model-testing basin in Washington.

Perhaps at no time was Mr. Connet more prolific in design than when on a train journey. He rarely returned without some new ideas on paper. On one short trip he removed his detachable cuff and sketched upon it the complete design fundamentals for the famous "Pull to Start, Pull to Stop" countershaft, which rapidly superseded the old wooden bar belt shifter. Thousands were sold.

About 1904 the company began the manufacture of the d'Auria high-duty pumping engine with its ingenious "loop liquid flywheel," which immediately attracted the attention of water-works men throughout the United States and Canada. The design details for these engines, built in capacities of from 1,000,000 to 10,000,000 gallons per day, were very largely Mr. Connet's.

Frequently he was called upon to devise some special machine tool for the shop. Once he made the complete designs for an electrically operated traveling crane of 35 tons capacity for the foundry, including a new and clever differential hoisting and trolleying control mechanism for which a patent was granted. This crane is still in service. His knowledge of shop methods enabled him to produce rapidly fully detailed drawings: these were always free-hand, but so neatly executed that they appeared to be mechanical.

No doubt Mr. Connet will be longest remembered as the inventor of the first registering device for use with Herschel's venturi tube. In 1892, with the collaboration of Walter W. Jackson, this highly ingenious machine was brought to successful completion. The invention earned the award of the John Scott Medal. The citation reads, "Its invention, design, and perfection are the fruits of great ingenuity and of much knowledge and painstaking labor—of vast benefit to the community by making of the venturi meter a practical working tool." A 36-in. venturi tube with the new register was exhibited at the Chicago World's Fair in 1893, measuring all the water supplied by the city. Curiously enough, this meter, which was regarded with no little skepticism, revealed an open branch valve and saved the exposition authorities some $25,000. The principle of integration is still in use by several instrument manufacturers. When telemetering seems now such a recent development, it is interesting to read in the first venturi catalog showing this register that "there may also be an electric device by which the record may be made at any distance (several miles if desired) from the meter." The invention of the register alone would entitle Mr. Connet to permanent recognition in the field of hydraulics, but it seemed to awaken new avenues in his fertile mind and there followed in continuous succession a great number of entirely new conceptions and improvements in flow metering and controlling devices. Some thirty patents were issued to him, among them the first venturi-filter effluent controller (now a requisite for all rapid sand filtration plants); an automatic reversible-flow venturi meter; sine integrator for differential instruments; mechanism for ship windlasses; automatic proportional flow regulator for two or more fluids (with Jackson and Gregory); vertical-spindle surface grinder; self-locking pump valve (with Richardson); automatic horsepower recorder for flowing fluids; multiple-pressure-chamber venturi tube; venturi meter for irrigation; distant signaling device; compound fire-service meter (with Graham); variable-speed device for automatically controlling chemical feeders; boiler feed-water regulator, floating-lever type; direct-acting flow controller (with Huxford); a meter for viscous oils. So well conceived and sturdily designed were his devices that many are still in daily use in municipal and industrial plants.

Of an insatiably inquiring mind, Mr. Connet was an inveterate reader on many subjects, particularly those of a scientific nature. The possessor of an accurate and retentive memory, a keen observer, a lover of fine music, deeply appreciative of the beauties and wonders of nature—these traits, plus a generous attitude toward others, made him a delightful companion. The privilege of associating with him in the practice of his chosen profession had all the attributes of a liberal education.

Mr. Connet had been a member of the A.S.M.E. since 1908, and also belonged to the Providence Engineering Society. He is survived by his widow, Esther (Robinson) Connet, whom he married in 1892, and by a son, Andrew.—[Memorial prepared by Charles G. Richardson, Providence, R.I. Mem. A.S.M.E.]

HERBERT RAY CONNOR (1886–1935)

Herbert Ray Connor, city clerk of Alameda, Calif., died in that city of peritonitis on May 7, 1935. He is survived by his widow, Susie Fontaine (Benton) Connor, whom he married in 1908, and by a son, Herbert Benton Connor.

Mr. Connor was born on May 8, 1886, at Virginia City, Nev., son of Samuel Bartlet and Virginia P. (Tilton) Connor. He prepared for college at the Worcester (Mass.) Academy, graduating in 1907. Four years later Brown University conferred an Sc.B. degree in mechanical engineering upon him. While at Brown he was elected to membership in Sigma Xi and Alpha Tau Omega.

Prior to entering college Mr. Connor had experience in drafting room and shop work and following his graduation he took a position as draftsman with the Meese & Gottfried Co., San Francisco. His work related chiefly to the design of elevating and conveying systems and power-transmission appliances and units. After about two years he was made sales engineer and he continued to serve the company in that capacity until 1920. He also acted as consulting engineer and was in charge of testing various types of equipment manufactured by the company and redesigning mechanism when necessary.

In 1920 Mr. Connor was chief engineer for the M. Phillips Co. and Phillips Milling Company, of San Francisco and Sacramento. He then opened an office in San Francisco for consulting work on elevating, conveying, and power-transmission problems, screening equipment and installation, and also some mining problems. He organized the California Rock Company in 1924, designed and erected its

plant at Pleasanton, Calif., and served the company as president for two years and subsequently as engineering salesman.

Leaving his duties with this company at the end of five years, he spent some time traveling with his family in Europe in 1930. In 1933 he served the Porcupine Mining Company, Haines, Alaska, as general manager. He was appointed city clerk of Alameda in 1935. For ten years prior to his death he was director of the Sixteen to One Mining Company, Alleghany, Calif.

Mr. Connor became a junior member of the A.S.M.E. in 1913 and was promoted to full membership in 1919. He also belonged to the Society of American Military Engineers and the National Rifle Association and was a Royal Arch Mason and Knight Templar.

JAMES CARR COOK (1879–1934)

James Carr Cook, vice-president and general manager of the J. B. McCrary Co., Atlanta, Ga., met death in an automobile accident while on a business trip to Florida on April 14, 1934. He was born at Cusseta, Ga., on April 14, 1879, the son of Mr. and Mrs. W. F. Cook. After securing a B.S. degree in electrical engineering from the Georgia School of Technology in 1903, he spent about a year as part owner and operator of a sawmill, then was employed for short periods by the Columbus Power Company, Columbus, Ga., Western Electric Company, New York, N.Y., and Bainbridge (Ga.) Water and Light Plant, where he was operating engineer.

In July, 1905, Mr. Cook became construction superintendent for J. B. McCrary, municipal engineer of Atlanta, Ga., and the following year was made field and office engineer for J. B. McCrary & Co., his duties covering surveys, investigations, designs, and estimates for water-works, electric-light, and sewer systems. In 1910, when the J. B. McCrary Co. was incorporated, he was elected its vice-president and chief engineer and given complete charge of all engineering work. He soon became general manager of the company and assistant to the president, in charge of new business, and during his long connection with the company distinguished himself as an executive and civil engineer.

Mr. Cook became a member of the A.S.M.E. in 1920. He also belonged to the American Society of Municipal Improvements and the American Society for Testing Materials. He is survived by his widow, Ermie (Stinson) Cook, whom he married in 1904, and by two sons, James Carr Cook, Jr., and John Cook, and by a daughter, Chastaine Cook.

LEON MILLARD CORNMAN (1900–1935)

Leon Millard Cornman, son of William Horace and Elizabeth (Lehigh) Cornman, was born at Carlisle, Pa., on November 3, 1900. He entered Carnegie Institute of Technology from the Carlisle High School and was graduated with a B.S. degree in 1924.

Soon after his graduation he entered the employ of the Newport News Shipbuilding & Dry Dock Co., and continued with that company until his death on February 20, 1935. He served successively in the engineering, production, inspection, and hydraulic departments. During recent years he had engaged in research work as assistant test engineer of the hydraulic laboratory.

Mr. Cornman became an associate-member of the A.S.M.E. in 1929 and was a first lieutenant in the Officers' Reserve Corps, U.S.A. He is survived by his widow, Miriam S. (Gildner) Cornman, whom he married in 1925, and by their two children, Elizabeth Mae and Leon Millard Cornman, Jr.

JAMES HOWELL CRARY (1886–1934)

James Howell Crary, son of William Proctor and Lillie (Howell) Crary, was born on July 13, 1886, in Brooklyn, N.Y., and received his early education there and at the Riverview Military Academy, Poughkeepsie, N.Y. He attended Amherst College for three years and then spent about a year each building motorboats at Westport, N.Y., and selling automobiles in New York, N.Y. From 1912 to 1915 he was employed in Bridgeport, Conn., by the Starbuck & Mattice Co., subsequently known as Mattice & Co., as automobile salesman, and also had charge of repair work. He then took the Reo automobile agency in Stamford, Conn., but in 1916 went with the Crucible Steel Company as inspector of shells for a time, and the following year took charge of the manufacture of superheater and boiler fittings for the Babcock & Wilcox Co., Bayonne, N.J. In October, 1917, following the entry of the United States into the World War, he entered the service of the Bureau of Steam Engineering of the United States Navy, as inspector of boilers and superheaters and their fittings.

Since the close of the War Mr. Crary had worked entirely on his inventions, especially on a two-cycle internal-combustion engine, on which he held patents in the United States and several other countries. He had also patented a hull construction for boats and a strainer for gasoline for automobiles, and had invented a spreading device for placing a net between an enemy submarine and a merchant vessel.

Mr. Crary became an associate member of the A.S.M.E. in 1918. His death on July 21, 1934, at Westport, N.Y., leaves a widow, Mercy H. (Lloyd) Crary, whom he married in 1917, and two sons, James Howell, Jr., and Bruce Lloyd Crary.

CHARLES SHARP CRAWFORD (1883–1935)

Charles Sharp Crawford, son of Edward Golay and Rose Hannah Crawford, was born on April 3, 1883, at Indianapolis, Ind. He died of pneumonia at St. Vincent's Hospital in that city on January 28, 1935, while on leave of absence from his position as chief engineer at the Opel Works, General Motors subsidiary in Germany.

Prior to his graduation from the Manual Training School of Washington University in 1902 Mr. Crawford served an apprenticeship in foundry and machine-shop practice. From 1902 to 1904 he had varied experience in pump repairing for the Dean Pump Works, electric-light installation and locomotive inspection for the Big Four Railroad, machinery installation for the Ceraline Mills, and in the toolroom of the Link Belt Company, all in Indianapolis. During the next two years he was chief draftsman, working on gas-engine and carburetor design for the Speed Changing Pulley Company in his home city, and after spending the fall of 1906 as draftsman in the experimental department of the Lozier Motor Company, Plattsburg, N.Y., he returned to the Speed Changing Pulley Company in the capacity of chief engineer of engine, carburetor, and chassis design.

In 1909 he became the first chief engineer of the Cole Motor Car Company, Indianapolis, with which he remained until 1916, with the exception of a few months during the latter part of 1910, when he was chief engineer and factory manager of the Westcott Motor Car Company, Richmond, Ind. He was made factory manager (in addition to chief engineer) of the Cole Motor Car Company in 1911 and assistant to the president the following year.

In 1916, he became associate chief engineer for the Premier Motor Corporation, Indianapolis, subsequently advancing to the positions of chief engineer, vice-president in charge of engineering, and director of the company.

From May, 1922, until 1928 he was chief engineer for the Stutz Motor Car Company. He left that organization to become affiliated with the General Motors Corporation, which sent him to Germany in 1929 to take charge of the Opel plant at Russelsheim. He had patented a number of inventions for the automotive industry and contributed to its publications.

Mr. Crawford became a member of the A.S.M.E. in 1918. He was a member of the Society of Automotive Engineers and served as a councilor of that society in 1918–1919. He was a charter member and former chairman of the Indiana Section of the S.A.E. He held the 32d degree in Masonry and was a Shriner. He belonged to the Columbia and Indianapolis Athletic Clubs.

Surviving Mr. Crawford are his widow, Adah E. (Williams) Crawford, and their daughter, Jane Alice Crawford.

CHARLES JACKSON DAVIDSON (1867–1935)

Charles Jackson Davidson, who was a manager of the A.S.M.E. for the term 1911 to 1914, died of pneumonia in Milwaukee, Wis., on May 26, 1935. He was born at Lanesboro, Minn., on July 6, 1867. the only son of Samuel Jackson and Harriet (Geer) Davidson, When he was four years old his father died and he moved with his mother to Germantown, Philadelphia, to live with his great-grandfather, Schuyler Geer, an importer of precious gems. After completing his early education he made a trip to the East Indies to buy pearls and returned with such an excellent collection that his grandfather wished to make him successor in his business. His inclination turned toward engineering, however, and he served an apprenticeship as machinist and steam fitter under Hubbard & Gere in Sioux City, Iowa, from 1885 to 1888 and passed the examinations of the Board of Education in Iowa the following year.

Mr. Davidson's first position was with the R. D. Fowler Packing Company at Sioux City, where he had charge of the boiler room at first and later was promoted to assistant engineer and in 1890 to chief engineer. During the five years he was located at this plant it was operated successively by R. D. Fowler, Ed. Haakinson & Co., and the Cudahy Packing Company.

In 1893 Mr. Davidson entered the employ of the Sioux City Traction Company as chief engineer. He resigned in 1899 to become chief engineer of power plants for the Milwaukee Electric Railway & Light Co., where he remained until 1911. During part of this period he

also served as consulting engineer for the Union Electric Light & Power Co., of St. Louis, Mo.

He designed and built a number of large power stations in Milwaukee and St. Louis and was recognized as an authority on steam engineering and power-plant design. In the spring of 1907 he made a trip abroad to study methods of handling street-railway problems in the various capitals of Europe.

Before resigning from the Milwaukee Electric Railway & Light Co. in 1911, Mr. Davidson designed and built the central heating system in Milwaukee and acted as general manager of that utility for a short time. He then went to Chicago to become vice-president of the firm of Woodmansee, Davidson & Sessions, Inc. (later known as the Woodmansee-Davidson Engineering Company), which engaged in electrical, mechanical, and hydraulic engineering, specializing in the design and construction of municipal power and light plants. This association continued until 1926, during which time Mr. Davidson invented the Moloch automatic steam-boiler stoker and an automatic water column.

In 1926 he purchased the controlling interest in the J. E. Gilbert Grinder Company of Milwaukee, of which he was president and general manager until his death.

He had published a booklet on "The Duty of the Engineer to Posterity."

In addition to serving as a manager of the A.S.M.E., of which he became a member in 1904, Mr. Davidson was a member of a special committee on a Bureau of Engineering Standards in 1913–1914, and of a Subcommittee (of the Committee on Meetings) on Depreciation and Obsolescence from 1913 to 1915.

He was one of the organizers and president in 1906–1907 of the Engineers' Society of Milwaukee, and was an associate member of the American Society of Naval Engineers. His clubs had included the Milwaukee Club, New York Engineers', Chicago Engineers, and the Hamilton Club, Chicago. He had attained the 32d degree in Masonry.

Surviving Mr. Davidson is his widow Ethel V. (Spence) Davidson, whom he married in 1901.—[Based, in part, on a biography compiled by the American Historical Society for the "Encyclopedia of American Biography."]

JOHN ALFRED DIXON (1867–1936)

John Alfred Dixon died at the Orange Memorial Hospital, Orange, N.J., on January 13, 1936, after a brief illness.

Mr. Dixon was born in East Orange, N.J., on May 26, 1867, the son of John S. and Phoebe Williams Dixon, both descendants of early Essex County settlers. He received his early education at the Ashland School in East Orange, and the Stevens Preparatory School. He was graduated from Stevens Institute of Technology, with the degree of mechanical engineer, in 1891.

While a student at Stevens he took an active part in the Stevens Engineering Society, and as an alumnus he served as first vice-president of the Alumni Association in 1913–1914, and as president in 1914–1915. He also served as an alumni trustee of Stevens.

Upon graduation from Stevens he entered the employ of the Pintsch Compressing Company, as assistant engineer, and spent several years in the construction, maintenance, and operation of Pintsch gas plants manufacturing oil gas for lighting railroad cars, gas buoys, and lighthouses. For some time he was superintendent of the Pintsch plant at Boston, Mass. In 1905 he was appointed general superintendent of the Pintsch Compressing Company, later becoming vice-president and general manager of the company, a subsidiary of the Safety Car Heating & Lighting Co. He became vice-president of the latter company in 1912, and president of the Pintsch Compressing Company in 1919.

He was vice-president of the Products Protection Corporation, another subsidiary of the Safety Car Heating & Lighting Co.

During his service with these various companies his activities were primarily concerned with engineering and manufacturing, and it was under his guidance that those organizations developed the lighting of cars by gas and electricity, the lighting of aids to navigation by gas, the cooling of freight cars for the transport of perishable products, the cooling of passenger cars for comfort, and the destruction of insect life in cereals and similar products.

Mr. Dixon was a charter member of the American Gas Light Association and the old American Gas Institute. At his death he was a member of the American Gas Association.

In 1913 he joined the Compressed Gas Manufacturers' Association, was elected to its Executive Board in 1920, and was a member of its Finance Committee for several years. In 1931 he was vice-president of that association. He was elected to membership in the A.S.M.E. in 1916.

He was for seventeen years a vestryman of Grace Episcopal Church of Orange, N.J. He was a member of the Essex County Country Club of Orange, the Downtown Athletic and the Railroad Club of New York, the Seabright Club, and the Quinnipiac Club of New Haven, Conn.

In 1898 he married Miss Charlotte Condit Perrine of Charleston, S.C., who survives him. He left two daughters, Mrs. Phyllis D. Ericson of Germantown, Pa., and Eloise, whose marriage to Mr. Francis Page Mackinney, of Essex Fells, N.J., took place on June 5, 1936, and a son, Alfred Brokaw Dixon, a student at Yale University. Mrs. Ericson has a son, John Eric, and a daughter, Linn Ericson.—[Memorial prepared by Geo. E. Hulse, New Haven, Conn., and Arthur P. Hagar, New York, N.Y., Members of the A.S.M.E.]

WALTER FRANK DIXON (1865–1935)

Walter Frank Dixon, who died of pneumonia on June 17, 1935, after a prolonged illness, was born in London, England, on June 23, 1865. His parents were William and Clara Dixon. He spent eight years at school in Bedford, England, the last three in a technical school.

At the age of 16 he went to New York, N.Y., where he served his apprenticeship from May, 1882, to June, 1885, in the drafting room and machine shop, and on the road, firing locomotives, of the New York, West Shore & Buffalo R.R. During this period he completed a night course in electrical engineering at Columbia University and in the summer of 1885 went to Europe to study locomotive practice.

From September, 1885, to January, 1886, he was employed as a draftsman for the West Shore Railroad at Frankfort, N.Y. During the next five months he served as a draftsman for the Barney & Smith Manufacturing Co., car builders, Dayton, Ohio. From July, 1886, to 1890 he served, first, as chief draftsman, with supervision of shopwork, for the Strong Locomotive Company, New York, then as draftsman and designer in charge of constructing a new foundry, boiler shops, power house, etc., for the Cooke Locomotive and Machine Works, Paterson, N.J.

In 1890 he went to the Rogers Locomotive Works, Paterson, N.J., as chief draftsman, later becoming a director of the company. His work with this firm was so outstanding that in 1895 he was sought by a locomotive syndicate to build the Sormovo Locomotive Works in Russia, where he served five years as manager and chief engineer.

This experience, his knowledge of the Russian language, and his ability to deal successfully with Russian labor, led to his selection, in 1900, by The Singer Manufacturing Company, to supervise the erection of a huge sewing machine factory at Podolsk, Russia. He often acted as arbitrator between opposing factions, enabling them quickly to settle their controversies.

He was appointed works manager of this plant, remaining in charge until 1917, when the factory was seized by the revolutionists during the overthrow of the Russian government. He and his family miraculously escaped death at the hands of snipers when with many other Americans they were driven from Russia, making a perilous trip through China to Japan, thence to San Francisco and New York.

He brought with him two imperial decorations, one the Order of St. Anna, presented by Czar Nicholas in recognition of his work in transforming the plant into a munition center during the early days of the war; the other, the Order of St. Stanislav, for his development of Russian commercial activities.

In 1918 he served as a director for the Russian Singer Manufacturing Company of Moscow, and as a special representative of the executive office of The Singer Manufacturing Company at New York.

During the years the United States was engaged in the World War, Mr. Dixon represented the company in the negotiation of war contracts, visiting Washington, D.C., frequently and aiding in the setting up of war work at the Elizabethport factory.

In 1920 he was appointed works manager of the company's Elizabethport plant, a post which he held until his death. During his long and efficient service as head of this factory, Mr. Dixon was held in the highest esteem by all of his associates.

In 1924 he was made vice-president of the Diehl Manufacturing Company, the electrical division of The Singer Manufacturing Company, manufacturing motors for sewing machines and other electrical devices.

He became a junior member of the A.S.M.E. in 1886 and a member in 1894. Active in Professional Division work, he served as chairman of the Standing Committee on Professional Divisions and as chairman of the Executive Committee of the Machine Shop Practice Division. Representing the Society, he served as chairman of the Sectional Committee on Standardization of Electric Motor Frame Dimensions. He was a past-president of the Engineers' Club of Plainfield, N.J., and in 1930 was a member of the regular nominating committee of the A.S.M.E. He had also been a member of the Institution of Mechanical Engineers since 1897. He was the author of

several papers read before various American engineering societies.

He was active in public affairs, being a director of the Elizabeth, N.J., Chamber of Commerce for several years. He served as a vice-president of the Chamber, a member of its National Legislative Committee, a delegate to the sessions of the United States Chamber of Commerce, and a member of the finance, membership, and industrial development committees. He was instrumental in forming the Manufacturers' Council, a subsidiary body of the Chamber, and his sound advice was helpful at the directors' sessions.

In 1923, Mr. Dixon was appointed to the Board of Directors of St. Elizabeth's Hospital, Elizabeth, and was largely responsible for the construction of a new wing to the hospital.

He served for several years as a director and member of the Board of Trustees of the Young Men's Christian Association of Elizabeth and contributed generously to the fund for the erection of a new building for which the Singer Manufacturing Company also made a large donation through his efforts. He was particularly interested in the industrial welfare work of the association, and was a member of the committees on industrial work, finance, and foreign service. He promoted clean outdoor recreation camps for boys at Waywayanda and Andover, N.J., and was the donor of many trophies for athletic events.

Prominent in many civic and philanthropic activities, Mr. Dixon gave unstintedly of his time, talent, and financial support.

When the Elizabeth Community Chest was formed in 1932, Mr. Dixon became a member of its managerial board and held the vice-presidency of the Board of Trustees until his death.

During the introduction of the National Recovery Act, he helped to organize the State industrial division of the N.R.A. He served on the Mayor's committee on free foreign trade zone.

He held a membership in the Singer Engineering Society, presiding at all of their meetings, and was a member of the Singer Club, Elizabeth Lodge of Elks, and Grace Episcopal Church of Plainfield.

Mr. Dixon was married twice, his first wife being Rosa Neale of Brooklyn, whom he married in 1886, and by whom he had one daughter, Dorothy, who survives him.

His second wife was Ludmila Beechevsky of Sormovo, Russia, whom he married in 1898. There were two sons, Vladimir and William, by the second marriage; Vladimir died in Paris in 1929, leaving a wife and son. Mr. Dixon's survivors, therefore, are a wife, a son, a daughter, and grandson.—[Memorial prepared by W. J. Peets, Elizabethport, N.J. Mem. A.S.M.E.]

LOUIS KARL DOELLING (1871–1935)

Louis Karl Doelling was born in Karlsruhe in Baden, Germany, on September 12, 1871, the son of Louis and Louise (Reuter) Doelling.

After the death of his father, his mother married Ernst Koerting, of Hanover.

He attended preparatory schools in Karlsruhe and Hanover, and then the Technical University of Hanover, where he received the degree of M.E. in 1890. Later he took a postgraduate course in Charlottenberg.

He served his apprenticeship with Koerting Bros. of Hanover, builders of engines, steam-heating and jet apparatus. Later he was draftsman for T. A. Maffai, locomotive builder in Munich. He then became a salesman, also testing engineer, for Koerting Bros. in Karlsruhe and Hanover. His last position in Europe was as manager of the Koerting Bros. Works in Vienna, Austria.

Ernst Koerting, of Koerting Bros., stepfather of Mr. Doelling, was a distinguished engineer and inventor of international fame. Many patents were granted to him, including those for the well-known Koerting injector and Koerting gas engines.

As a result of the training and experience Mr. Doelling received while with Koerting Bros., he became a conspicuous expert and authority on internal-combustion engines.

In 1905 Mr. Koerting purchased control of the De La Vergne Machine Company, then located at the Foot of East 138th Street, New York, and sent Mr. Doelling over to take the management of the engineering and manufacturing departments of the company which in addition to refrigerating machinery, was also building internal-combustion engines, including the Koerting gas engines. Later Mr. Doelling became vice-president and then president of the company.

In 1917 the United States Government purchased the De La Vergne Machine Company, and in 1920 the Wm. Cramp & Sons Ship & Engine Building Co., of Philadelphia, bought it from the Government. Mr. Doelling was retained by the Cramp Company as vice-president and general manager of the De La Vergne Machine Company.

In 1928 the plant was removed to Philadelphia and operated as the De La Vergne Division of the Cramp-Morris Industrials, by whom Mr. Doelling was employed as consulting engineer until 1930, when he resigned and became vice-president of the U. S. Fire Protection Company, of Hoboken, N.J., which position he held up to the time of his death.

Mr. Doelling was a very efficient shop manager and systematizer. He made many improvements in manufacturing processes in the plant of the De La Vergne Machine Company. One of his outstanding traits was his humane and kindly treatment of employees. He was greatly respected and esteemed by all the workmen, and took a special interest in their welfare and social life. His benevolence and consideration were personally extended to a large number of employees and they were greatly benefited by the interest he took in their well-being and that of their families.

Mr. Doelling became a member of the A.S.M.E. in 1906. In 1908 he joined the American Society of Refrigerating Engineers, and was its president in 1915.

He died at his home in Mountain Lakes, N.J., on June 16, 1935, and is survived by his widow, Emma (Renck) Doelling, whom he married in 1898, and by two sons, Hans and Klaus Doelling.—[Memorial prepared by Louis Baron, New York, N.Y., for many years closely associated with Mr. Doelling at the De La Vergne Machine Company.]

JOSEPH ESPLIN DORWARD (1890–1934)

Joseph Esplin Dorward, whose death occurred on September 22, 1934, was born in San Francisco, Calif., on October 31, 1890. He attended an evening technical school there for three years while serving an apprenticeship as a machinist at the United Engineering Works. After working in the drafting office of the same company on marine-engine design for about two years he entered upon a period of sea service with the Pacific Mail Company, Pacific Coast Steamship Company, Union Oil Company, Matson Navigation Company, and Alaska Packers Association. He continued in this work until April, 1918, with the exception of a year with the United Engineering Works, installing boilers and machinery, and about a year and a half with the General Petroleum Company as chief inspector of reconditioned vessels.

From April, 1918, to September, 1919, Mr. Dorward served in the United States Navy, part of the time as engineer watch officer and part as 1st assistant engineer, with the grade of senior lieutenant, on the U.S.S. *Agamemnon*, from New York to Brest. He returned to civilian life as chief estimator for the Lord Dry Dock Construction Company and Lord Construction Company, New York, and held that position until the end of 1921. His next position was that of manager of the North River Branch of the Morse Dry Dock & Repair Co., New York, where he worked for about two years, and from then until 1928 he was inspector for the engineering department of the United States Lines, serving on the S. S. *Leviathan*.

Mr. Dorward entered the employ of The Export Steamship Corporation on January 1, 1928, as superintendent engineer. His duties included the supervision of the engine-room staff and all mechanical appliances aboard the vessels of the corporation and the placing and supervision of repairs. In September, 1928, he was promoted to the position of operating manager in charge of all operations afloat and the appointment and supervision of the personnel of the ships. He also had general supervision of the activities of the port captain, port steward, and port engineer. He continued in these duties until his death.

Mr. Dorward became a member of the A.S.M.E. in 1923 and also belonged to the Society of Naval Architects and Marine Engineers. He is survived by his widow, Lucille May Dorward.

THOMAS FAWCETT DuPUY (1875–1936)

Thomas Fawcett DuPuy, consultant in plant equipment methods and production, died suddenly on March 21, 1936. He was born at Athens, Ohio, on April 9, 1875, son of Thomas Fawcett and Ann Agnes (Donly) DuPuy. His high-school education, secured at Pittsfield, Mass., was followed by private tutoring and practical mechanical training in his father's woolen mills at Hinsdale, Mass. His father was an authority on textiles and an inventor of equipment in this field.

Mr. DuPuy's work as plant consultant began in 1908 for the National Acme Company, of Cleveland. After several years there he took a position as sales engineer for the Windsor (Vt.) Machine Company, which offered him the opportunity to serve also as consultant to manufacturers of automatic screw machines, turret lathes and related machinery and tools. He made surveys of many plants, advising as to valuation as well as improvements in equipment, layout, and methods looking toward increasing the quality and quantity of production at reduced costs.

After the outbreak of the World War in 1914 he was called upon by many munitions manufacturers for advice as to equipment and manu-

facturing methods, and during the early part of 1917 he was consulted by the Ordnance Department in connection with ordnance equipment. Later in the year he was commissioned a captain in the Aviation Section of the Air Corps, assigned to the Aviation Repair Depot at Love Field, Dallas, Texas, as supply and disbursing officer. He was required to organize and equip the depot, which handled repairs for several fields.

After leaving the service in February, 1919, Mr. DuPuy was works manager for about two years for J. H. Williams & Co., Buffalo, N.Y., manufacturers of special drop forgings and forging dies, and a standard line of wrenches. From 1921 to 1924 he was vice-president and general manager of the Canton Forge & Axle Co., Canton, Ohio, and during the next five years engineer for the American Machine & Foundry Co., Brooklyn, N.Y. Much of his work during the latter period related to special automatic packaging machinery for tobacco, cigarettes, candy, textiles, and other products.

In 1929 Mr. DuPuy became chief purchasing engineer for the Amtorg Trading Corporation, New York. He specified and purchased machine tools and special machinery for various industries, including tractors and turbines, automotive and electrical equipment, and woodworking machinery. Since 1932 he had been engaged largely in work of the Professional Engineers Committee on Unemployment, in New York.

Mr. DuPuy became a member of the A.S.M.E. in 1921. He is survived by his widow, Blanche Eloise (Brooks) DuPuy, whom he married in 1908.

CARL EHRMANN (1882–1935)

Carl Ehrmann, whose death occurred on August 13, 1935, was born at Fuerth, Bavaria, Germany, on November 23, 1882. He was the son of Simon and Mathilde Ehrmann.

Mr. Ehrmann was educated in Germany, securing a degree in mechanical engineering from the University at Munich in 1906. His engineering experience in Germany included an apprenticeship in the foundry and machine shops of the Vereinigte Maschinenfabriken Nuernberg Augsburg and three and one-half years in drafting and layout work and estimates for coal- and coke-handling plants with Koelnische Maschinenbau A.G., later known as Berlin Anhaltische Maschinenbau A.G., at Koeln-Bayenthal, Germany.

Mr. Ehrmann came to the United States in 1910 and secured employment with the Didier-March Company of New York and South Bethlehem, Pa., working on drafting and calculations in connection with by-product coke-oven plants. He was made chief draftsman in March, 1912, and remained with the company until the close of that year.

During the next two years Mr. Ehrmann engaged in consulting practice in New York. From 1914 to 1916 he was chief engineer for the Curtis Bay Chemical Company, Baltimore, Md., and then went to Tulsa, Okla., to serve the Oklahoma Petroleum & Gasoline Co. as consulting and chief engineer. He then went into his own consulting business, building many oil refineries and gasoline plants. Since 1919 he had been a partner in the Valuation Company of America, New York, engaged in consulting and appraisal work.

Mr. Ehrmann became a junior member of the A.S.M.E. in 1911 and a member in 1913. He was also a member of the American Association for the Advancement of Science and of the Verein deutscher Ingenieur.

Surviving Mr. Ehrmann are his widow, Beatrice (Schuster) Ehrmann, whom he married in 1916, and two children, Charles and Margaret Ehrmann.

HARVEY FELDMEIER (1871–1935)

Harvey Feldmeier, chief engineer of the Cherry-Burrell Corporation, Little Falls, N.Y., died of heart disease on September 19, 1935, while in Troy, N.Y., on business.

Mr. Feldmeier was a native of Brooklyn, N.Y., son of Max and Elvire (D'Asnoy) Feldmeier. He was born on August 27, 1871, and was graduated from the Brooklyn Technical Institute in the class of 1890, with the degree of B.S. He then entered Rensselaer Polytechnic Institute at Troy, where he received his degree of civil engineer in 1892. He was a member of the college fraternity of Delta Kappa Epsilon and of the Sigma Xi honorary society.

Following his graduation Mr. Feldmeier secured work as an engineer for the New York State Canals. He was put in charge of the construction of a dam across the Mohawk River at Little Falls. He went to work for D. H. Burrell & Company, manufacturers of dairy machinery, in December, 1892, and had devoted his abilities and faithful interest continuously to the problems and the advancement of the business and the dairy industry since that time. When D. H. Burrell & Company merged with other companies, Mr. Feldmeier was made chief engineer of the Cherry-Burrell Corporation.

Mr. Feldmeier had taken out more than fifty patents on dairy apparatus and was regarded as one of the ablest engineers in the dairy-machine industry in this country.

Mr. Feldmeier had been a member of the Little Falls Board of Public Works since 1912 and was its vice-president at the time of his death. He was a Democrat in politics but never sought elective office. However, he was alternate delegate to the Democratic national convention in Baltimore in 1912, and alternate delegate to the national convention of the party in Houston, Texas, in 1928, from the 33d congressional district. He was also one of the presidential electors in 1928.

He was a commissioner of the General Herkimer home, a position to which he was appointed in 1931 by Governor Roosevelt.

He became a member of the A.S.M.E. in 1931 and belonged to the Rensselaer Society of Engineers.

Mr. Feldmeier was twice married, his first wife being Miss Helen Mitchell, of Little Falls, who died many years ago. In 1915, he married Miss Lela B. Lumley of Utica, who survives with four children, Allan L., Elizabeth Jane, Robert Harvey, and Edward Burrell Feldmeier.

CHARLES HORACE FESSENDEN (1885–1934)

Charles Horace Fessenden, professor of mechanical engineering at the University of Michigan, Ann Arbor, Mich., died on July 26, 1934. He became a member of the faculty in the fall of 1908 as instructor in mechanical engineering. He was made an assistant professor in 1912 and a professor in 1919.

Professor Fessenden was born in St. Louis, Mo., on January 28, 1885, a son of Timothy Dwight and Mary J. (Snively) Fessenden. He attended the St. Louis Manual Training School for three years and secured his B.S. degree in mechanical engineering at the University of Missouri in 1906. He spent that summer in the Engineering Department of the New York Telephone Company, and in the fall became a draftsman for the Babcock & Wilcox Boiler Co. at Barberton, Ohio. In the spring of the following year he was made checker and assistant chief draftsman, and he remained with the company until October, 1908. During the two years following his graduation he was also carrying on post-graduate work at the University of Missouri, which conferred an M.E. degree upon him in June, 1908.

During the years 1906–1926 Professor Fessenden assisted Dean M. E. Cooley and others in the valuation of many public-utility properties—electric, gas, electric railway, and steam railway—and since 1919 had served as consulting engineer on a number of public-utility projects. In 1917–1919 he served in the Ordnance Department, U.S.A., first with the rank of captain and later as major, stationed at the Frankford Arsenal, Philadelphia, Pa. He was the author of a book on "Valve Gears," published in 1915.

Professor Fessenden became a junior member of the A.S.M.E. in 1909 and a member in 1917. He also belonged to the American Society of Refrigerating Engineers and the Society for the Promotion of Engineering Education, as well as to Alpha Tau Omega fraternity, several Masonic organizations, and a number of clubs.

Professor J. E. Emswiler, long associated with Professor Fessenden on the faculty of the University of Michigan, has written the following tribute to him:

"It was as a teacher of engineering students that Professor Fessenden's greatest service was rendered to his profession. He took into the classroom not only a rich and ordered program for the hour, but a creed of precision and accuracy that communicated itself to the class. He was a stern though understanding taskmaster whose lessons were sometimes painful to take; but among returning alumni, many of his best friends were those who had grumbled most as undergraduates. Time and experience had proved to them that the continued effort required in Professor Fessenden's classes was the finest kind of training for the tasks of life."

MARTIN FEYBUSCH (1884–1935)

Martin Feybusch, a native of Germany but later a naturalized citizen of the United States, was born at Stettin on April 17, 1884. After completing his preparatory education he served an apprenticeship with steam engine and boiler manufacturers in Stettin, and worked for a time as electrician at a power house there. He was graduated from college at Friedberg in 1904, had an engineering position in Hamburg the following year, working on steam engines, and then spent several years in Paris. There he was engaged in drafting for automobile manufacturers.

Returning to Germany, he served for a time as chief draftsman and manager for the Maschinenbau-Gesellschaft at Ronsdorf and subsequently at the Jagenberg Maschinenwerke in Düsseldorf. He came

to the United States as representative of that company in 1913, serving as vice-president and treasurer of the Jagenberg Machine Co., Inc., New York.

In 1918 he founded the New Jersey Machine Corporation, Hoboken, N.J., of which he was president at the time of his death on October 25, 1935. He held many patents on glue applying equipment.

Mr. Feybusch became a member of the A.S.M.E. in 1914. He belonged to the Liederkranz in New York. Surviving him are his widow, Martha Feybusch, and three children, Margerite, Marcelle, and Martin Feybusch, Jr., all of Weehawken, N.J.

JOHN MILTON FOSTER (1888–1935)

John Milton Foster, associate professor of aeronautical engineering at North Carolina State College of Agriculture and Engineering, died in Raleigh on February 14, 1935, at the age of 46.

Professor Foster was born in Richmond, Ky., on May 20, 1888, the son of John Milton and Elizabeth (White) Foster. After a secondary schooling in his native city he attended the University of Kentucky and received the degree of bachelor of mechanical engineering in 1911. This institution also conferred upon him the degree of mechanical engineer in 1923.

After graduation from the University in 1911, Professor Foster entered the special apprentice course of the Allis-Chalmers Company in Milwaukee, Wis., and was employed for a time in the various departments of this plant. In 1912 he went to Nashville, Tenn., where he took a high-school position as teacher of industrial arts. In 1918 he went to Raleigh as assistant professor of mechanical engineering at North Carolina State College, and in 1924 was promoted to the grade of associate professor. More recently he was specially designated to fill the position which he held at the time of his death.

Professor Foster was one of the pioneer aeronautical instructors of the Southeast, having served as a licensed ground school instructor for the Curtiss-Wright Flying Service in Raleigh prior to the development of aeronautical courses at North Carolina State College. At the college he was in charge of the Aeronautical Option in the Department of Mechanical Engineering from the time of its inception in 1929. He did considerable flying, although he was not a licensed pilot.

In addition to teaching, Professor Foster found time to engage in a considerable amount of professional work. During the summer of 1918 he was associated with the George A. Fuller Company as assistant to the chief mechanical engineer, designing a power house for a wood alcohol plant at Lyles, Tenn.; and in 1919 he designed a water works plant for the town of Zebulon, N.C., under the direction of W. M. Piatt, a consulting engineer of Durham, N.C. In later years Professor Foster was registered as a professional engineer in the state of North Carolina and did considerable work in the field of machine design, especially in connection with inventions and patents.

Professor Foster was well known in engineering circles throughout the Southeast. In addition to his affiliation with the A.S.M.E., of which he became a member in 1921, he held membership in the Society for the Promotion of Engineering Education, the North Carolina Society of Engineers, the Raleigh Engineers Club, and the Aeronautical Chamber of Commerce of America. He was also a member of the Alpha Tau Omega fraternity and was a charter member and a past-president of the Lions Club of Raleigh. He served two terms as chairman, and briefly as secretary-treasurer, of the Raleigh Section of the A.S.M.E., and at the time of his death was secretary-treasurer of the North Carolina Society of Engineers and secretary of the North Carolina Engineering Council. He was a member of the 1934 Nominating Committee of the A.S.M.E. from Group IV, and served as secretary of this committee. He was also a member of the Committee on Aeronautical Engineering Education of the Aeronautical Chamber of Commerce. He was an active church member.

Surviving Professor Foster are his widow, the former Miss Bessie Schnell of Nashville, whom he married in 1915, and their four children, Albert White, Virginia Duncan, John Milton, Jr., and Elizabeth S. Foster.—[Memorial prepared by Robert P. Kolb, University of Alabama; formerly a member of the faculty at North Carolina State College. Mem. A.S.M.E.]

KARL E. GARLING (1888–1935)

Karl E. Garling, whose death occurred at Bernardsville, N.J., on October 26, 1935, was born at Seneca Falls, N.Y., on September 11, 1888. At the age of twelve he began an apprenticeship with the Seneca Falls Manufacturing Company, upon the completion of which he was employed there as a journeyman machinist until 1906. During the last four years of this time he also took evening instruction in machine design under the designer for the company.

From 1906 to 1912 Mr. Garling held various positions in machine shops, drafting rooms, and printing plants in Auburn, Rochester, and Buffalo, N.Y., and Chicago, Ill., part of the time in charge of departments. Thereafter, until 1919, he was connected with the Whitlock Printing Press Company, Derby, Conn., except for the year 1914–1915, when he was with the Carey Printing Company, New York. His work for these companies included the design, erection, and maintenance of machinery.

In 1919 Mr. Garling opened his own office at Newark, N.J., and devoted himself to the design of special machinery, chiefly for paper specialties. In 1928 his business was merged with the Murray Sales Corporation, of Newark, which later became the Garr Manufacturing Company, and then the Soda Straw Manufacturing Company. About two years before the death of Mr. Garling the business was sold to the Berst-Forster-Dixfield Company, New York. Mr. Garling was the inventor of a gas burner for drying newsprint, and soda straw machinery.

Mr. Garling became a member of the A.S.M.E. in 1929 and belonged to the Masonic fraternity. His wife died in 1934.

DANA GOVE GRIFFIN (1873–1935)

Dana Gove Griffin, who died at Fort Worth, Texas, on February 26, 1935, had for many years been sales engineer and representative at the Fort Worth branch of the Bruce-Macbeth Engine Company, of Cleveland, Ohio. He had been located in Texas since the spring of 1909, serving for more than two years as general superintendent of the Fort Worth city water works and from the beginning of 1912 until the fall of 1922 as manager of the engineering department of the Hardwicke-Etter Company of Sherman. His connection with the Bruce-Macbeth company began in 1922.

Regarding his early work in Fort Worth, T. J. Powell, commissioner of water works in that city, wrote in 1911:

"Mr. Griffin was superintendent of the Fort Worth Water Works for more than two years, during my first term of office as water commissioner. He took hold of a run-down plant with inadequate machinery and a system of artesian wells that were yielding about a million gallons of water per day. During the time he was with the department he increased the output of the same wells to seven million gallons per day and transformed the various plants of the city, while they were in continuous operation, into a first-class system. This work was done during the years 1909–1911, a period of the greatest drought ever known in this state, and owing to his genius in developing the wells the city was saved from a disastrous water famine because the surface supply which had been used before his time utterly failed during these years.

"I regard Mr. Griffin as the best well expert and pneumatic engineer I have ever met. He can bring more water out of a well with compressed air than any man I have ever known, and it gives me great pleasure to commend him to any one desiring the services of such an expert."

Mr. Griffin was born at Valdosta, Ga., on October 13, 1873, son of Ivan L. and Laura Eugenia (Keller) Griffin. He completed his academic education at the Quitman, Brooks Co., Ga., Academy in 1889. Early in 1893 he went to Kansas, becoming a machinist apprentice in the locomotive department of the Chicago, Rock Island & Pacific Railway at the Horton shops. He returned to Valdosta in 1895 to become chief engineer and superintendent of the city water works, a position which he held for about eleven years. A new plant was designed and installed by him during this period.

From the beginning of 1907 until he went to Texas in 1909 Mr. Griffin was connected with the pneumatic engineering department of the Ingersoll-Rand Company, working chiefly at the Tarrytown, N.Y., and Phillipsburg, N.J., plants. He was also in charge for a time of the air-lift system for men working under high pressure below water on the Hudson River tunnel, seeing that they were properly brought out to normal air pressure, passing through air chambers in which the pressure was gradually reduced. While this is now done by automatically controlled chambers, the control of pressure to prevent "bends" was manual at the time the Hudson tunnel was constructed.

Mr. Griffin was elected an honorary life member of the National Ice Manufacturing and Refrigerating Engineers' Association in 1914. He became a member of the A.S.M.E. in 1929. He also belonged to the National Geographic Society, the Masonic fraternity, and the Shriners, and was a deacon in the Broadway Baptist church of Fort Worth. Hunting was his favorite recreation, and from time to time, as he found the leisure, he hunted deer and other large game in southern and southwestern Texas.

Surviving Mr. Griffin are his widow, Aline (Shaw) Griffin, whom he married in 1895, and two children, Dana G. Griffin, Jr., and Margaret Louise (Mrs. Wilbur B.) Duke.—[Based, in part, on a biography compiled by the American Historical Society for the "Encyclopedia of American Biography."]

ERIC EDWIN HAGBERG (1875–1935)

Eric Edwin Hagberg, for more than forty years associated with the International Harvester Company, died at St. Luke's Hospital in Chicago, Ill., on September 28, 1935.

Mr. Hagberg began his long connection with the International Harvester Company as an apprentice toolmaker in 1892, at the McCormick Works in Chicago. Later he advanced to special machine and tool designing, and from 1908 to 1911 served as master mechanic. During the next six years he was factory manager at the Neuss Works of the company in Germany. He returned to the United States to organize an Efficiency Department at the McCormick Works and to take charge of all Government work there. Following the War he was stationed at the Neuss Works, until 1924. After another short period at the McCormick Works he was transferred to the Canton (Illinois) works, to take the position of superintendent. Since 1928 he had been located at the Picayune Works in Mississippi in a similar capacity.

Mr. Hagberg was born at Highwood, Ill., on September 3, 1875, son of Eric and Lovisa (Erikson) Hagberg. To supplement his grammar school education he attended evening classes at the high school and at Lewis Institute in Chicago. Married in 1903, he is survived by his widow, Fidelia (Johnson) Hagberg, and by their two children, Eric Edwin, Jr., and Dorothy Fidelia.

Mr. Hagberg became a member of the A.S.M.E. in 1920. He was a member of the Odd Fellows Organization in Canton and was president of the Rotary Club there in 1927–1928. He was president of the Rotary Club in Picayune in 1931–1932, and had twice received the Silver Loving Cup of that club, presented for best living up to Rotary ideals. He was also active in Boy Scout and Y.M.C.A. work in that city, serving as president of the Board of Directors of the latter.

DWIGHT KIMBALL HALL (1882–1934)

Dwight Kimball Hall, a member of the firm of Frank A. Hall & Sons, New York, N.Y., and general manager of its factory at Southfields, N.Y., died at his home in Goshen, N.Y., on December 25, 1934. He was a native of Brooklyn, N.Y., where he was born on January 16, 1882, son of Francis Augustus and Mary (Randall) Hall. He entered Stevens Institute of Technology from the Lawrenceville (N.J.) Preparatory School and was graduated with an M.E. degree in 1908.

During his college vacations Mr. Hall worked in the Hall plant and following his graduation he had charge of rearranging and installing machinery there. He then became assistant superintendent of the foundry, engaging in the manufacture of brass and iron bedsteads, tables, and other furniture. He was made a member of the firm in 1915.

Mr. Hall became a junior member of the A.S.M.E. in 1910. He had served on the Board of Education of Goshen, and belonged to the Blooming Grove (N.J.) Club. He was fond of hunting and fishing.

Surviving Mr. Hall are his mother and his widow, Florence (Baldwin) Hall, whom he married in 1925.

FREDERICK ARTHUR HALSEY (1856–1935)

Designer, inventor, journalist, and author—Frederick Arthur Halsey was each and all of these to a degree to make him distinguished for assured achievement in engineering. As designer he contributed to the development of the rock drill and other compressed-air machinery; as an inventor he initiated one of the earliest mechanisms of industrial management; as a journalist he was editor of one of America's leading technical magazines; as an author he wrote numerous professional papers and books of recognized authority.

Halsey was of American Colonial ancestry, belonging to the ninth generation of his family resident in this country. From this heritage he was a member of the Order of Founders and Patriots of America, Sons of the American Revolution, and Pilgrims. His life was of generous length, 79 years. Born in Unadilla, N.Y., on July 12, 1856, he died in New York, N.Y., October 20, 1935.

Halsey obtained his technical education, and received his engineering degree from Cornell University in 1878. He studied under Dr. John E. Sweet and became one of "Sweet's Boys," that group of loyal followers of a great leader and educator.

After a year's work as machinist in his home town, he secured his first engineering position in charge of the testing room of the Telegraph Supply Company, later the Brush Electric Company of Cleveland, Ohio. This connection was short, from the autumn of 1879 to the spring of 1880. Then followed another short engagement as draftsman with the Delamater Iron Works in New York. His real life's work began in 1880 when he became engineer for the Rand Drill Company of New York, a connection that continued for fourteen years, or until 1894. During the last four years of this engagement Halsey was in charge of the plant of the Canadian Rand Drill Company, Sherbrooke, Que., Canada.

While with the Rand Company Halsey worked out two of his notable achievements: the "Slugger" rock drill, and the Premium Plan of wage payment. His versatility and pioneer spirit are shown by these two developments, the one a mechanical invention, the other a management innovation. The latter brought him the greater recognition. A few months before his death he wrote of the beginning of his Premium Plan in these words:

"The Premium Plan of paying for labor was an outgrowth of the labor disturbances of the 1880's, a period which saw the development of the militant organization, the Knights of Labor, accompanied by an epidemic of strikes. These conditions were accompanied by an extended discussion of means for harmonizing the relations of Capital and Labor, one of which was the Premium Plan, which, at the beginning, was not markedly successful as the plan received the unqualified condemnation of Samuel Gompers.

"I was at that time the mechanical engineer of the Rand Drill Company, and I urged a trial of the plan upon the company, but without success for reasons that it would be useless to recall. The Canadian Rand Drill Company was organized in 1880, and of it I was offered the position of general manager. The conception of the Premium Plan was then several years old, but I had felt that a publication of it, as a mere project, would accomplish nothing. It was, however, my hobby and, seeing before me a free hand to give it and some mechanical ideas a trial, I accepted the position for that purpose."

Halsey's Premium Plan is the earliest of the incentive plans for paying for labor. It is a constant sharing plan with a time guarantee. A standard time is set from records of past performance, and the time the workman saves is divided between him and his employer, the former usually receiving from one quarter to one half of the saving. At the same time the regular wage at time rate is guaranteed. The plan is simple to introduce, emphasizes the time saved, and has an excellent effect on the thinking of the workman. For many years after Halsey gave his plan to his brother engineers it was the most widely used of American incentive wage plans.

In 1923 Halsey was the recipient of the A.S.M.E. Medal, "awarded to Frederick Arthur Halsey for the Premium System of wage payments."

In 1894 Halsey made the business connection that was to be the longest and last of his life. He then became associate editor of the *American Machinist;* in 1907 he was promoted to editor in chief, and in 1911 became editor emeritus on the occasion of his retirement. His contributions to machinery building during these seventeen years were notable, particularly in the field of machine design. Several of his studies passed over from magazine articles to permanent literature in book form.

During his years as editor in chief of the *American Machinist*, that publication made a new record for circulation and influence. Halsey strengthened its position as a medium for the presentation of information on machine design, as a forum for the discussion of management and economic problems, and as a news paper for the new developments in machinery building, both American and foreign. An examination of the issues for the years 1907 to 1911 shows that some four out of five numbers have a feature article devoted to a striking development in the field of the magazine. The pages devoted to the airplane are evidence of this feature of Halsey's editorial policy. His editorials in defense of that great invention, and his replies to skeptics, particularly those in foreign countries, are masterpieces of vigorous technical discussion.

While Halsey was associate editor the adoption of the metric system became a public issue in the United States. Shortly after the opening of the present century, the Committee on Coinage of the House of Representatives reported favorably out of committee a bill for the adoption of the metric system of weights and measures. At once, Halsey became one of the most active opponents of the bill, writing and speaking against it. At the annual meeting of the A.S.M.E. in 1902 he presented a strong argument against the proposed change in a paper entitled, "The Metric System." The session was one of the most controversial ever held by the Society.

Acting upon resolutions passed at sessions following the one at which Mr. Halsey presented his paper, the Council authorized the appointment of a committee to summarize the arguments for and against the adoption of the metric system; and the distribution of a letter ballot to the members of the Society to obtain an expression of their sentiment on the subject.

The committee report was published with the Halsey paper and the voluminous discussion on it in Volume 24 (1903) of Transactions. The paper and discussion occupied more than two hundred pages,

and the committee report, presenting the pro-metric arguments and anti-metric replies and additional appendixes contributed by the anti-metric members of the committee, took another eighty pages. The committee was agreed that legislation designed to compel the exclusive use of the metric system was not desirable, and the report on the letter ballots sent out from the secretary's office showed a similar feeling on the part of a heavy majority of the individual members who voted.

The outcome of the metric fight was the defeat of the bill in Congress. In recognition and appreciation of Halsey's activities in the struggle the National Association of Manufacturers presented him with a beautiful cathedral chime hall clock.

After his retirement from his editorship Halsey organized and became the first commissioner of the American Institute of Weights and Measures. This work was a continuation of his metric fight. During this period also, he produced his most notable book, the "Handbook for Machine Designers and Draftsmen." The activities of the closing years of his life were mainly travel and promoting Anglo-American relations.

Among the books of which he was author are these: Slide Valve Gears; Locomotive Link Motion; Slide Rule; Worm and Spiral Gearings; Metric Fallacy; Design and Construction of Cams; Method of Machine Shop Work; Metric System in Export Trade; Weights and Measures in Latin America; and the handbook previously mentioned.

Very few memorials to engineers refer to social accomplishments. Halsey's record permits an exception of this general rule, for he was a finished ballroom dancer, and kept up this artistry to the close of his life. In his statement of Biographical and Professional Data, prepared for the A.S.M.E. in mid-summer, 1935, he wrote with pride, "They do say I can still dance."

Halsey loved books, and was a book collector. In his will he left several treasures to the A.S.M.E. Included in this gift are: Reuleaux's "Kinematics of Machinery;" Willis' "Principles of Mechanics;" Davies' "Metric System;" and "The Essays of Benjamin Thompson, Count of Rumford."—[Memorial prepared by L. P. Alford, New York, N.Y. Past Vice-President, A.S.M.E.]

ALFRED EMIL HAMMER (1858–1935)

Alfred Emil Hammer, son of Thorvald Frederik and Delphina (Lundsteen) Hammer, was born in Boston, Mass., on March 8, 1858, and died in Branford, Conn., on May 9, 1935. His father, a native of Copenhagen, was graduated from the Royal School of Navigation in Denmark, took up his residence in Boston in 1842, and later spent a few years at sea and in Cuba, where he erected and managed a sugar mill.

In 1864, with a brother, Emil C. Hammer, and others, he purchased the plant of the Totoket Company, which had been struggling with the manufacture of malleable-iron castings at Branford since its inception in 1845. A new company was organized, the plant equipped with special machinery for the manufacture of pipe fittings, and its name changed to Malleable Iron Fittings Company.

Alfred Hammer was educated in the public schools of Branford, and at Russell's Military Academy in New Haven. Just at this time a Swedish chemist, Ernst Sjöstedt, pupil of Professor Eggerts of the University of Upsala, Sweden, had been engaged to install a chemical laboratory in the plant of the Malleable Iron Fittings Company, one of the first in the industry. Here the young Alfred Hammer got his first glimpse of the possibilities of chemistry applied to the problems of the foundry. Fascinated, he decided against a formal college course, to apply himself to the solution of some of those problems, the importance of which to the foundry operations he had already divined. A few months' work with Sjöstedt, a few more under Professor George J. Brush at Yale College, gave him the methods and technique for his explorations. For exploration it was; foundry problems were solved by rule of thumb, fractures were the only guides to the quality of irons, pig iron was bought from this furnace or that because it gave the results sought, no one knew why. Certain charcoal blast furnaces were reputed to furnish iron of superior quality for the malleable-iron foundry and brought a premium in the market.

Young Hammer, however, was not easily satisfied; he insisted on knowing the reason why. A long series of analyses of raw material and finished product, with careful checking of results in the laboratory and in the foundry, and he had the answer, the relation of the elements, carbon, silicon, manganese, and sulphur and their effect. Charcoal pig iron from this or the other furnace was no longer a name to conjure with; the analysis told the story, and it is believed that it is to Alfred Hammer that credit is due for this important contribution to our metallurgical knowledge.

From the beginning, Mr. Hammer was engaged in the malleable-iron industry uninterruptedly and actively during his entire lifetime. Upon his father's death in 1901, he assumed the management of the business, becoming treasurer and general manager. In 1921 he was made president, continuing in all three offices until his death.

Mr. Hammer gave freely of his time to public affairs, served a term in the Connecticut Legislature in 1889–1890 as representative, and in 1905–1906 as state senator, in 1920 was a presidential elector on the Republican ticket, and was president of the Republican Club of Branford. Upon his father's death he succeeded him as treasurer of the James Blackstone Memorial Library of Branford, was president from 1925 until his death, and was secretary and treasurer of the Henry Whitfield State Historical Museum at Guilford, Conn. He was a director of the Second National Bank of New Haven, Conn.; and a trustee of the Connecticut Savings Bank of New Haven, Conn., and of the Branford Savings Bank. He became a member of the A.S.M.E. in 1884, and received its fifty-year membership medal in 1934. He was a member also of the American Institute of Mining and Metallurgical Engineers, American Society for Testing Materials, American Association for the Advancement of Science, New Haven Colony Historical Society, Connecticut Historical Society, National Institute of Social Sciences, Society for the Preservation of New England Antiquities, Connecticut Manufacturers Association, and of the Union League, Quinnipiack, Lawn, and Country Clubs of New Haven, and the Pine Orchard Country Club of Pine Orchard, Conn.

His outside interests many, Mr. Hammer took especial pleasure in the study of the early history of Branford and its neighborhood, and of the minerals of Connecticut, of which he made a noteworthy collection.

Reserved but not distant, dignified but approachable, responsive, kindly, and quick with his interest and sympathy, Mr. Hammer was helpful and generous in all good causes, contributing freely but so unobtrusively that few were aware of the extent of his benefactions.

He married twice, first on September 27, 1887, Cornelia Hannah Foster of New Haven, their children being Forrester Lundsteen of Branford, Rosalind Cornelia (Mrs. Charles F. Clise of Seattle, Wash.), Delphina Lundsteen (Mrs. Henry M. Clark, Jr., of Suffield, Conn.) and Thorvald Frederick of Branford. On June 10, 1905, he married Rosamond Swan of Brookline, Mass., who, with his four children, survives him.—[Memorial prepared by A. H. Jameson, Malleable Iron Fittings Company, Branford, Conn.]

JOHN HAYS HAMMOND (1855–1936)

John Hays Hammond, mining engineer, was born in San Francisco on March 31, 1855. His father, Richard Pindell Hammond, was a graduate of West Point and had served with distinction in the Mexican War. He had been ordered to the new territory of California for garrison duty, and arrived at San Francisco on the steamer *Oregon* on April 1, 1849. He was on the ground even before the gold rush of '49. His duties were not onerous. He was caught in the surge about him and resigned his commission in 1851, was elected to the state assembly in 1852 and chosen speaker.

That same year, 1852, Colonel John Hays, an old Texas Ranger and also an officer in the Mexican War, arrived by vessel from Panama, with his sister Sarah Lea, a widow with a small daughter. Soon after her arrival she met Richard Hammond and they were married in 1854. John Hays Hammond, the oldest of three brothers, was born in March of the following year.

Hammond's father was one of the leaders in the pioneer life of California, with interests all over the state. The boy went to school in San Francisco, Oakland, and the preparatory department of the University of California. In his vacations he hunted and camped up and down the Sierras and as far as the sleepy Mexican town of Los Angeles and panned gold for fun in the Chinamen's diggings. When he was sixteen his father decided that he should go east to school, and he started off for Washington on the new transcontinental railway opened two years before. He had a year at the Hopkins Grammar School in New Haven and entered the Sheffield Scientific School of Yale University in 1873. His career at Yale, he says, "was a pleasant mélange of books, professors, comradeships, social diversions and athletics." He was never very good, nor very bad. As with many others, the chief impression made upon him came from the personalities of some of his teachers, whose friendship he carried through life. Upon his graduation in 1876 he went to Germany and studied for three years at Freiberg, the foremost school of mining engineering in the world at that time.

When he returned to California in 1879 he found his technical training an actual handicap, but he was no stranger to frontier life and wisely kept Freiberg in the background. It was not many years before his superiority over "practical miners" became so evident that apologies were no longer necessary.

After a few months as an assayer he joined the U. S. Geological Survey on mine examinations which took him to practically all the

gold-mining operations on the Coast. When he had finished this work he went east and married, on January 1, 1881, Natalie Harris, whom he had met in Germany a few years before. His first fee of five hundred dollars was received for a mine examination which he made upon his return. During 1881 he was profitably employed on examinations of various properties in California, Nevada, and Arizona and as consulting engineer for a powder company.

Early in 1882 he went to Mexico as manager of a mine at Minas Nuevas, near Agiovampo on the west coast. Seven months later his wife and baby joined him, but had to return north on account of the roughness and dangers of the life. He himself left soon after, little richer, except in valuable experience.

In 1883 Hammond opened an office in New York as a consulting mining engineer. His practice grew rapidly and took him to Mexico. Central and South America, and throughout the West. The following year he was made consulting engineer for the Empire and North Star mines in California and a little later of the Bunker Hill and Sullivan Mine in the Coeur d'Alene district of Northern Idaho. He became part owner and president of the latter company, which proved immensely valuable, producing, up to 1932, ore valued at more than $167,000,000.

While in England in 1893 Hammond received an offer from Barney Barnato to become engineer of his properties in South Africa, which he accepted. He went to Johannesburg, taking his family and a group of young engineers as his staff. After a thorough inspection of all the properties Hammond made various important recommendations, but Barnato, whose attention was centered on the stock market, failed to avail himself of the counsel. Feeling himself in a growingly intolerable position, Hammond resigned at the end of six months. The separation was friendly and the two men remained on good terms until Barnato's death. In fact, Barnato stood bail for Hammond for $100,000 at the time of his trial and remained in Pretoria for six months, doing everything in his power for him.

As soon as Hammond's resignation became known, Cecil Rhodes wired him to come to Cape Town. Rhodes asked Hammond to take charge of all his mining interests on the Rand and told him to name his own salary. Hammond named $75,000 and participation in the profits. "All right," said Rhodes. "But," Hammond added, "I want to deal directly with you, without interference from your local directors. Unless this can be arranged I can't accept." Rhodes picked up a piece of paper and wrote on it "Mr. Hammond is authorized to make any purchases for going ahead, and has full authority, provided he informs me of it and gets no protest." In this manner Hammond became consulting engineer of the Consolidated Gold Fields of South Africa, and soon afterward of the British South Africa Company, and on the strength of this scrap of paper spent many hundred thousand pounds.

Perhaps Hammond's greatest professional triumph was the development of the "deep level" mines. The vein at the Rand has a very heavy dip, something in the order of 60 deg. All the mines then operating were outcrop mines, worked from the surface, with the deepest workings about 500 ft down. Hammond told Rhodes that in his opinion the vein and its values held down for at least several thousand feet, and that property beyond the existing claims, then only farm land, could be bought up and mined as profitably as the existing claims. This involved working at depths of approximately 5000 ft. Aside from the geological gamble there was serious question among engineers as to whether mines could be worked at such a depth. Rhodes staked his fortune on Hammond's judgment, sold out the holdings of the Consolidated Gold Fields in the existing mines, and utilized the capital so raised to buy the new property and develop deep-level mines. It was a tremendous undertaking, involving vast expenditures, not only before a cent could be returned, but before success could even be assured. It was a triumph for Hammond that the veins were picked up within a few days of the time, and a few feet of the location, which he predicted, and these deep level mines were a complete success. One of them, the Robinson Deep, has produced $160,000,000 and is operating at a depth of 8500 ft at a lower cost than the outcrop properties in 1894.

The reform movement of the Uitlanders and the Jameson Raid are matters of general history. It was forced by the reactionary and oppressive policies of Paul Kruger and a small group about him. The whole movement was ruined by the precipitateness of Jameson in entering Transvaal before the plans were ready. The revolt failed and Hammond, with three others, was arrested, tried, and condemned to be hanged. As the result of heavy pressure on the Transvaal government, his sentence was commuted to imprisonment, then he and the others were released on payment of smashing fines, and as soon as he could close up his affairs he left South Africa permanently, except for several visits.

He arrived at Plymouth in July, 1896, took a house near London, and for the next four years practiced as a general consultant, dividing his time between work for the Rhodes interests, with trips to South Africa once or twice a year, and in building up a clientele in London. In 1898 he made an extensive examination and report on the mineral resources of Russia and Siberia. Twelve years later he made a similar trip, but the rush of political changes defeated his recommendations. Hammond was now world famed and his activities brought him into contact with all the great financial and mining interests in Europe.

Increasingly, however, opportunities were opening up at home, and in December, 1899, he returned to New York and reopened an office there after an absence of seven years. He was soon busy with important commissions in Mexico, Colorado, and Nevada. He was a pioneer in developing the gold-dredging operations along the Yuba River in California, which handle profitably gravel running as low as seven cents per cubic yard. In 1903 Hammond became consulting engineer and general manager for the Guggenheim Exploration Company on a salary and an interest in the profits, a basis similar to the one he had had with Cecil Rhodes. They were active in the development in many properties under his leadership. Conspicuous among these was the Utah Copper Company, which has produced nearly $600,000,000 worth of copper, gold, and silver, and has paid over $185,000,000 in dividends. This company was a pioneer in the development of low-grade porphyry ores. All told, these ores have produced, throughout the United States and Latin America, up to 1931 inclusive, an output of $2,871,300,000. In 1907 Hammond resigned his connection with the Guggenheims on account of ill health. He always considered his association with them as one of the most satisfactory periods in his whole career.

Although he was never again in active practice, his many interests continued to claim his attention. Among these were irrigation projects, water supplies, hydroelectric and oil developments. He built a beautiful home in Washington, D.C., and another in Gloucester, Mass., and for nearly thirty years he was able to enjoy his success and utilize its possibilities and influence for public service.

Mr. Hammond was a lecturer at Harvard, Columbia, and Johns Hopkins, gave the Hammond Mining and Metallurgical Laboratory to Yale, and for a time was professor of mining engineering there. He was special ambassador for the United States to the coronation of King George V in 1911, president of the Panama Pacific Exposition Commission to Europe in 1912, chairman of the World Court Commission, 1914–1915, chairman of the United States Coal Commission, 1922–1923, and president of the National League Republican Club. He was a fellow of the American Association for the Advancement of Science and the American Academy of Arts and Sciences, president of the American Institute of Mining and Metallurgical Engineers in 1907–1908, and a member of the A.S.M.E. since 1910 and of many clubs in New York, Washington, San Francisco, Denver, Salt Lake City, and London.

He was a contributor to the magazines and scientific journals, and in 1935 he published a two-volume autobiography which was one of the most successful books of the year. It is of absorbing interest, for few men have had such a story to tell, and gives intimate pictures of famous men who have made history on three continents during the past forty years.

He was keenly interested in many civic movements, such as the Y.M.C.A. and Boy Scouts, but he was especially identified with the Boy's Clubs of America, of which he was an active supporter and vice-president for the last fifteen years of his life.

He died suddenly of heart failure on June 8, 1936, in the study of his home in Gloucester. His wife died in 1931. Their children were Harris, John Hays, Richard Pindell, and Natalie Hays. John Hays, Jr., is himself a distinguished inventor in many fields.

John Hays Hammond lived long and lived vividly. He made friends and kept them. He was one of the great mining engineers of all time, a splendid citizen, and an example of the best type of American engineer.—[Memorial prepared by J. W. Roe, New York, N.Y., Mem. A.S.M.E.]

D. ROBERTS HARPER, 3d (1885–1935)

D. Roberts Harper, 3d, was born at Ridley Park, Pa., on June 17, 1885, the son of Daniel Roberts Harper, Jr., and Blanche G. (McIntire) Harper. He prepared for college at the Philadelphia Manual Training High School and was graduated from the University of Pennsylvania with a B.S. degree in 1905. He was elected to membership in both Sigma Xi and Phi Beta Kappa. Postgraduate work led to Ph.D. in 1910. He was a Hector Tyndale Fellow in 1907–1908 and an instructor in physics at the University from 1904 to 1909.

Beginning as an assistant physicist at the United States Bureau of Standards in 1909, Mr. Harper devoted about fifteen years to research work at the Bureau's Division of Heat and Thermometry, largely on the subjects of refrigeration, thermodynamics, automotive power-

plant tests, and fuels. He held the position of physicist when he left in 1925 to study problems of refrigeration and the cooling of electrical machinery for the General Electric Company, at Schenectady. He remained there until 1930 and during the last four years of that time served also as associate professor of physics at Union college.

When the Coal Research Laboratory was being organized at the Carnegie Institute of Technology in 1930, Mr. Harper joined the staff as physicist. He continued in that position until 1934, since then he had been connected with the Federal Emergency Relief Administration at Washington, D.C.

Mr. Harper became a member of the A.S.M.E. in 1930. He was a Fellow of the American Association for the Advancement of Science and also of the American Physical Society; a past director of the American Society of Refrigerating Engineers; and a member of the Society of Automotive Engineers, Washington Academy of Sciences, Washington Philosophical Society, and the Société Francaise de Physique. He had served on committees on the Standardization of Refrigerators and the Safety Code for Mechanical Refrigeration of the American Standards Association.

Surviving Mr. Harper, whose death occurred on October 19, 1935, are his widow, the former Miss Mary Elma Lewis, of Baltimore, Md., whom he married in 1911, and one daughter, Edith Lewis Harper.

GRENVILLE A. HARRIS (1876–1934)

Grenville A. Harris, whose death occurred on January 31, 1934, following an illness of six months, was born at Elizabeth, N.J., on December 24, 1876, the son of Conover S. and Sarah J. Harris. He entered the Centenary Collegiate Institute at Hackettstown, N.J., in 1891 and remained there for two years. Subsequently he attended the Stevens Preparatory School, Hoboken, N.J. During the summer vacations of his school years he worked as fireman, oiler, and engineer on a steam yacht and he continued in similar work for something over a year after leaving school. In October, 1897, he was put in charge of the erection and operation of engines and boilers for the Clark Wire Works, Elizabeth, N.J., but in April of the following year entered the Spanish-American War, serving as chief gunner's mate and electrician on the U.S.S. *Vixen* until the close of the year. He was awarded a war medal for his services during the engagement of July 3, 1898.

After taking the student course of the Westinghouse Electric & Manufacturing Co. in 1898, Mr. Harris became construction engineer for the company and had charge of the erection of a number of important plants. In April, 1901, he was appointed resident engineer in New York, N.Y., in charge of the electrical and mechanical work of Takata and Company, a Japanese importing and exporting company, agents of the Westinghouse organization for Japan. He continued in this position until 1916.

During the next five years Mr. Harris was chief engineer for the American Steel Export Company of Delaware, in charge of the construction of steel plants. When the company was liquidated and a new company formed in 1921, he left to enter business for himself, representing several manufacturers in export activities.

About ten years prior to his death Mr. Harris became export engineer, with an office in New York, for The Black-Clawson Company of Hamilton, Ohio, manufacturers of paper- and pulp-mill machinery. The company regarded him as exceptionally competent to handle exports of heavy machinery.

Mr. Harris became a member of the A.S.M.E. in 1907. He was an associate of the American Institute of Electrical Engineers and also belonged to the Japan Society and American Asiatic Society and to the Machinery Club, New York. He had been a member of the Board of Fire Commissioners of Elizabeth, N.J., for sixteen years, and prior to that time had served the Board in an advisory capacity at various times. He married Miss Blanche W. Sheffer, of Baltimore, Md., in 1901, and is survived by her and by two children, Grenville C. Harris and Althea M. (Mrs. Charles G.) Black, and a grandchild, Charles G. Black, 3d.

JAMES JOEL HART (1905–1935)

James Joel Hart was killed in an automobile accident at Hicksville, L.I., N.Y., on May 22, 1935. He was born on September 15, 1905, at Union Springs, Ala., the son of James Joel and Lola (Frazier) Hart. He attended high school in Montgomery, Ala., and in 1925 went to New York, where he entered the employ of the Bell Research Laboratories as a mechanic's helper. He also entered evening school at Cooper Union and in 1930 received his B.S. degree in mechanical engineering.

Of his later work at the Laboratories, the *Bell Laboratories Record* for June, 1935, states:

"In 1928 he became a technical assistant in the Chemical Laboratories, working in the precious-metal group of the metallurgical section. Here he was engaged in the experimental development and production of filament cores for vacuum tubes. Mr. Hart became an expert in the process of rolling very fine filaments, particularly those made from nickel alloys and platinum and assisted in the design of a new filament rolling mill used in this work."

Mr. Hart is survived by his widow, Catherine Virginia (Drewry) Hart, whom he married in 1931, and by a son, James Joel.

He became a junior member of the A.S.M.E. in 1930.

CLAUDE HARTFORD (1885-1936)

Claude Hartford, who had been associated for twenty years with the New York Steam Corporation, died at his home in Maplewood, N.J., on February 4, 1936, after a long illness. He was widely known in the real estate and professional engineering fields in the Greater New York area. As a consulting engineer he was recognized as a specialist in the heating of large buildings in Manhattan south of Fifty-Ninth Street, and he also engaged in the application of engineering to passenger elevators, as vice-president of the American Elevator Company, and to welding, as secretary of the Wilson Welding & Metals Corp.

Mr. Hartford was born in Brooklyn, N.Y., on October 1, 1885, a son of William and Catherine (Halloran) Hartford. He was graduated from Public School No. 41, in Brooklyn, in 1900 and from the Commercial High School there three years later. He then attended Stevens Preparatory School for one year and Stevens Institute of Technology for two years before entering Sibley College at Cornell University, from which he was graduated as a mechanical engineer in 1910.

Mr. Hartford's early work was performed in Brooklyn with the Cross, Austin & Ireland Lumber Co., laying out a blower system and machinery for a new mill; the Soss Manufacturing Company, as chief draftsman, designing and directing the manufacture of dies and pressure casting machinery; and the Transit Development Company, as assistant engineer of distribution, testing and inspecting materials and making tests at substations and on the lines of Brooklyn street railways. He went to New London, Conn., in April, 1912, as engineer for the Connecticut Turbine Manufacturing Company, and was engaged in designing and testing turbines there until he became assistant to the chief engineer of the New York Steam Company in July, 1913.

Mr. Hartford became a junior member of the A.S.M.E. in 1912 and a member in 1917. He was also a member of the New Jersey Society of Professional Engineers and a licensed professional engineer both in New York and in New Jersey. He was one of the organizers of the Omega Alpha Pi high-school fraternity at the Commercial High School and was an active member of the Phi Sigma Kappa college fraternity. He belonged to the Cornell Club of New York.

Surviving Mr. Hartford are his widow, Lillian Fullerton Hartford, two children, Claudia Fullerton and Harry Richard Hartford, and his brothers Ernest and Lionel Hartford.

IRVING N. HAUGHTON (1879–1935)

Irving N. Haughton, president of the Haughton Elevator & Machine Co., of Toledo, Ohio, died at his home in that city on March 21, 1935.

Mr. Haughton was born in Toledo on April 23, 1879, son of Nathaniel and Frances Caroline (Bush) Haughton. He completed his high school education in 1896 and during the next two years was car clerk for the Cincinnati, Hamilton & Dayton R.R. During the remainder of his life he was connected with the Haughton business.

The Haughton Foundry & Machine Co. was started in 1867 by Mr. Haughton's father. They did a general foundry and machine business and worked into the manufacture of farm implements and steam engines and started building elevators (hydraulic, belt power, and hand elevators) about 1885. The business was reorganized in 1897 under the management of Mr. Haughton's eldest brother, who had been directing the business for some time.

Mr. Haughton started work with the new company in 1898. After a short period of training, he entered the selling end of the business, which grew rapidly from this time on, largely through Mr. Haughton's energy and ability. He became sales manager and secretary and continued in that capacity until the death of his brother in 1915. The company was then reorganized with Mr. Haughton as president, and a new plant was built to take care of the growing business. He remained its active manager and the guiding spirit of the business until his death.

Mr. Haughton had been a member of the A.S.M.E. since 1924. He was president for three terms of the Elevator Manufacturers

Association of the United States and represented that association on the Sectional Committee on a Safety Code for Elevators. He had also served as its representative on the Sectional Committee on Standardization of Elevators, which was discharged in 1929. He served on many other committees of the Association, particularly on the Labor Committee, of which he was a member for 15 years. He assisted greatly in the formation of the Ohio State elevator code, acting as chairman of the committee. He also served on committees in the Toledo Commerce Club and the Toledo Merchants and Manufacturers Association.

He was a lieutenant in the Naval Reserve in 1910–1912, and a delegate of the United States to the International Committee of the Building Industry and Public Works in 1925.

He had attained the 32d degree of Masonry and was a Shriner. He belonged to the Toledo Club for years and served as trustee and in various activities of the club. He was greatly interested in Rotary and took an active part in charitable work, particularly that for crippled children. He was a member of Inverness Golf Club for about twenty-five years and devoted considerable time to its development and growth.

He enjoyed golf greatly and played a great deal. He liked to fish and camp in the woods. He also enjoyed music keenly, particularly grand opera. He was a constant reader, particularly of history, biographies, scientific developments, the development of boats and ships, and mystery and detective stories.

Mr. Haughton is survived by his widow, Frances G. (Gates) Haughton.

JOHN VIRGIL HENDERSON (1886–1935)

John Virgil Henderson, whose death occurred at Wyandotte, Mich., on March 4, 1935, was born at West Lafayette, Ind., on July 12,1886. His parents, Albert Wilson and Anna (Songer) Henderson, survive him, as do also his widow, F. Edna (Lemon) Henderson, whom he married in 1922, and a sister, Ruth Songer Henderson, a teacher in Lafayette, Ind.

Mr. Henderson was graduated from Purdue University with a B.S. degree in mechanical engineering in 1909 and three years later secured his M.E. degree. During this period he served an apprenticeship on electrical power equipment with the Westinghouse Electric & Manufacturing Co., Pittsburgh, Pa., and worked successively on power maintenance for the American Locomotive Works there and on design and erection for the Western Electric Company, Chicago, Ill. Subsequently he was employed by the Chicago, Burlington & Quincy R.R. and the Chicago, Milwaukee & St. Paul R.R., on design, operation, and inspection.

After a period of consulting power-plant practice in Chicago, Mr. Henderson went to Detroit, where he had a shop for making tools, continued his consulting practice, and served as chief engineer for the Hotel Statler. He was also munitions inspector for the French Commission for part of 1915 and was foreman of the Ford Eagle shipbuilding plant from 1917 to 1919.

In November, 1919, Mr. Henderson became lubricating salesman for the Standard Oil Company, and he continued in that position until December, 1930, calling on the large industrial accounts of the company. From then until about a week before his death, when he had taken a position with the Detroit Packing Company, as chief engineer, he was engaged in refrigeration and engine-room maintenance.

Mr. Henderson became an associate-member of the A.S.M.E. in 1919. He had attained the 32d degree in the Masonic fraternity, and was a Shriner and Knight Templar. He also belonged to the Odd Fellows and Knights of Pythias.

AARON S. HERSHEY (1907–1935)

Aaron S. Hershey, vice-president and production manager of Hershey Metal Products, Inc., Derby, Conn., died on January 15, 1935. He was born at Hanover, Pa., on September 8, 1907, son of Paul H. and Mary J. R. (Bair) Hershey. He attended the Shelton High School, at Shelton, Conn., and the Choate School in Wallingford, and entered the Sheffield Scientific School of Yale University with the class of 1929.

After receiving his B.S. degree he was employed until the end of 1930 at the Hawthorne Works of the Western Electric Company, Chicago, Ill., working on the introduction of new machine-tool equipment.

Since then he had been associated with the Hershey Company, of which his father is president. The company was reorganized in July, 1934, its name being changed from the Hershey Metal Products Company to Hershey Metal Products, Inc., and it was at that time Mr. Hershey became vice-president and production manager. He had previously served as engineer and production manager and had handled the novelty division of the company since its inception early in 1932.

Mr. Hershey became a junior member of the A.S.M.E. in 1932.

REINHARD ADOLPH HISS (1903–1935)

Reinhard Adolph Hiss died at Hammond, Ind., on February 17, 1935, from chill and exposure when an iceboat on which he was riding overturned in water.

Born on August 28, 1903, at Barberton, Ohio, a son of Reinhard Simon and Louise (Scherzer) Hiss, he attended the Barberton grammar and high schools. His work at the University of Michigan, which he entered in 1921, led to the degree of bachelor of science in engineering (mechanical engineering) in 1925 and election to membership in Tau Beta Pi. Following his graduation he became a member of a graduate engineer's training course known as The Central Station Institute, operated by utility companies under the management of Samuel Insull. The course consisted of four three-month assignments in various engineering departments.

At the end of this year of special training Mr. Hiss took a position with the Commonwealth Edison Company in Chicago as assistant to the efficiency engineer. In September, 1930, he became superintendent of the efficiency division of the Chicago District Electric Generating Corporation. He was also on the Board of Governors of the State Line Club at the State Line Station, where he was located.

Mr. Hiss became a junior member of the A.S.M.E. in 1929. He was very fond of music and active in church and Boy Scout work. Surviving him are his widow, Amina (Calhoun) Hiss, whom he married in 1928, and two children, Reinhard C. and Amina Suzanne Hiss.

RALPH W. HOWE (1885–1935)

Ralph W. Howe, identified with the petroleum industry since his graduation from Cornell University, died as a result of injuries in an automobile accident at Dallas, Texas, on October 23, 1935. He had been a member of the A.S.M.E. since 1932.

Mr. Howe was vice-president and general manager of the Atlantic Pipe Line Company, of Dallas, at the time of his death. He had been with the company since its organization in 1928. His positions previous to that time were with the Tuscarora Pipe Line Company, the Standard Oil Companies of New Jersey and Louisiana, and the Hope Natural Gas Company, from 1908 to 1910; The Atlantic Refining Company, Philadelphia, Pa., 1911 to 1922; and Atlantic Oil Producing Company, Shreveport, La., 1922 to 1927. His experience with these companies covered the design, construction, and operation of pipe lines and the operation of refineries.

Mr. Howe was a member of the American Petroleum Institute and was active for many years in the standardization program and other activities of the Production Division of the Institute. He was a member of the Committee on the Standardization of Oil Country Tubular Goods from May, 1928, up to the time of his death. In August, 1931, he was appointed national chairman of the Committee on the Standardization of Steel Tanks for Oil Storage, which office he also held until his death. During this administration as chairman of the Tank Committee, he inaugurated the preparation of a specification on all-welded steel tanks, which was adopted shortly after his death. He was recognized as an outstanding authority in his profession, and his counsel was continually sought in the affairs of the Production Division.

Mr. Howe was born at South Salem, N.Y., on August 22, 1885, son of E. W. and Emily J. Howe. He prepared for College at Blair Academy and received his M.E. from Cornell in 1908. He is survived by his widow, the former Miss Isabella D. Betzner, of Hornell, N.Y., whom he married in 1912, and by two children, Ralph W., II, and Jean Marie Howe.

EDWARD PIERCE HULSE (1870–1936)

Edward Pierce Hulse, Chairman of the Graphic Arts Division of the A.S.M.E. and a member of the board of the Graphic Arts Research Bureau, died in New York, N.Y., on April 3, 1936.

Mr. Hulse was born in Chicago, Ill., on November 22, 1870, the son of Parmenas Brown and Katherine (Smith) Hulse. After receiving his early education in public and private schools, he was sent with his brother, in 1884, to the Alexander Military Institute at White Plains, N.Y., where they prepared for a college course in civil engineering, which his brother subsequently pursued at Cornell University. Edward, however, was urged by his father to take a classical course at college, and being unwilling to do this he went to work as a reporter for the *Akron Beacon* in 1888.

Newpaper work was Mr. Hulse's chief occupation from then until

1902. He was a substitute telegraph editor for the *Pittsburgh Dispatch* in 1890; on the *New York World* in 1892; with the *Jacksonville Citizen* in 1893; *Atlanta Constitution in* 1895; *Pittsburgh Dispatch*, again, in 1899, in various capacities; managing editor of the *Pittsburgh Herald* in 1901; and a stockholder, co-organizer, and telegraph editor of the *Atlanta News* in 1902.

He then took up passenger traffic work in Atlanta for the Georgia Railway & Electric Co. In 1903 he became traffic agent for the New England Street Railway Lines, Boston, Mass., where he remained until 1906, when he was connected for a short time with the Traffic Department of the New York, New Haven & Hartford R.R. During these years he contributed a good many articles on transportation problems to the technical press.

In 1907 Mr. Hulse returned to the newspaper field as desk editor of the *New York American*, and the following year became news editor of the American Press Association.

About this time Mr. Hulse's interest in engineering evidenced itself in his organization of the Emerson Street Sweeper Company, with franchises in a number of cities. A few years later his interest in the Imperial Electric Motor Company (owners of the Berg-Ledwinka patents on the four-motor four-wheel drive), led to the organization of the Electro-Coach Corporation for the manufacture of buses and taxicabs. As manager and traction expert for the company he laid out bus routes for the City of New York which were basic lines for many years. Still later he became treasurer of the Randall Motor Corporation, New York.

In 1910 Mr. Hulse's printing and engineering interests merged in the efficiency engineering firm of Crittenden and Hulse, New York, which was devoted for many years to practical mechanical work in printing plants, handling contracts in printing and publication houses throughout the East and South, and in Canada. His technical articles dealing with printing plant problems which were published during this period included "Planning Buildings to Suit Printing Plants," "Layout of Cylinder Pressroom to Avoid Vibration in Structure," and "The Offset Process—A Practical Treatise," a guidebook in use for many years.

As a printing plant consultant Mr. Hulse had been of considerable assistance to John A. Hill in connection with his Pearl Street plant and during the planning of the new building at 36th Street and Tenth Avenue. In October, 1914, Mr. Hill called upon him to make a study of his five engineering publications, and the manner of presenting articles to the public. The result of his study, a 320-page "Hill Engineering Weeklies Manual," intended as a guide to all engineering writers, was in process of publication at the time of Mr. Hill's death in January, 1916. Mr. Hulse, who held the copyright for the manual, resigned his position upon the death of Mr. Hill, and their plans for the manual were not carried out.

During the next few years Mr. Hulse devoted himself to such war activities as Liberty Loan campaigns and other money-raising and patriotic drives. He was also, beginning in June, 1917, national director of publicity and information for the Boy Scouts of America, and organized publicity for the Eastern Department of the National War Work Council. He aided in organizing the nation-wide search for standing black walnut for gun stocks and airplane propellers, secured large quantities of peach pits for gas-mask charcoal, and supplied the Government with much useful information on other subjects. His work for the Boy Scouts led to his selection in 1920 for the position of international secretary of the Big Brother Movement, which he declined because of the large amount of traveling required. He did serve temporarily, however, in connection with work in the New York area.

Of all his associations perhaps the one which most completely satisfied his inclinations was that with the American Type Founders Company, Jersey City, which began in 1921. As manager of their Merchandise Department Mr. Hulse improved many of the old items, devised and patented new ones, including the Universal quoin key and the hacksaw pressroom knife. At the same time he was carrying on experiments in the practical application of stainless steel in this field, using it successfully for type-high and plate gages, type scales and line measures, composing and make-up rules, ejector press punches, and many other devices. In 1925 he became president of the Stainless Steel Manufacturing Company, New York.

Mr. Hulse became a member of the A.S.M.E. in 1925 through his interest in the Printing Industries Division of the Society (renamed the Graphic Arts Division in 1935). He helped to develop the series of conferences in the printing industry sponsored by the Division, and gave much of his time and effort to the committee work of the Division in the capacity of its chairman in 1929–1930, secretary from 1932 through 1935, and as chairman again during the months immediately preceding his death. He was instrumental also in the organization of the Graphic Arts Research Bureau in 1935.

From 1928 to 1934 Mr. Hulse was a member of the editorial staff of the A.S.M.E., where his varied experience formed a valuable background for his work on technical papers for *Mechanical Engineering* and the TRANSACTIONS and his newspaper training and familiarity with the mechanical details of printing were frequently of great help. He was editor of "Arc Welding," published in 1929 and comprising papers submitted to the A.S.M.E. for the prizes offered by the Lincoln Electric Company, Cleveland, in 1928.

During the last two years of his life Mr. Hulse struggled against his failing health, unwilling to give up participation in those activities which had long claimed his interest; saddened also by the death in 1933 of his wife, the former Miss Nellie B. Griffin of Jersey City, whom he married in 1909.

Prior to the death of Mrs. Hulse, they made their home in Westfield, N.J. He became interested in the history of the State of New Jersey and was a joint editor of "New Jersey: Life, Industries, and Resources of a Great State," published by the New Jersey State Chamber of Commerce, in 1928.

Mr. Hulse is survived by a sister, Mrs. Cornelia L. (Edward W.) Preston, of Bridgeport, Conn., and a nephew, Bennett Preston.

FRED CROCKER HUSSEY (1894–1934)

Fred Crocker Hussey, who became a member of the A.S.M.E. in 1927 and served as secretary of the Kansas City Section of the Society in 1932-1933, was born on September 10, 1894, at Carthage, Mo. He entered the University of Missouri from the Carthage High School and was graduated with a B.S. degree in engineering in 1918. During the summers of 1915 and 1916 he worked in Manitowoc, Wis., at the shipyards there and at the Richards Iron Works, and he spent the winter of 1917–1918 at the Atlas (Mo.) Powder Mills. For eight months in 1919 he was employed by the American Car & Foundry Co., Madison, Ill.

For about seven years, beginning in the fall of 1919, Mr. Hussey was connected with the United Iron Works, Inc., in Kansas City, rising from the position of draftsman and estimator to that of manager of the Pomona Pump Department. Part of his early work was done at the plant at Iola, Kan., and part at Joplin, Mo., where he was in charge of several draftsmen working on the design of electric hoists and mining cars. Later he helped to design a Pomona double-stroke deep-well power pump, established sales agencies, and developed advertising for the Pomona pump.

Since July, 1926, he had been with the Kansas City Water Department, beginning as draftsman and serving as mechanical engineer for the department at the time of his death the latter part of August, 1934.

Mr. Hussey is survived by his widow, Mrs. Ruth Hussey, a son, William, and a brother, Frank Hussey.

GENTARO IGARASHI (1875–1935)

Gentaro Igarashi, director of the Ajikawa Iron Works, Osaka, Japan, a member of the A.S.M.E. since 1921, died on February 2, 1935.

Mr. Igarashi was born at Utsunomiya, Japan, on May 1, 1875. He attended the College of Science and Engineering of Kyoto Imperial University, receiving a degree in mechanical engineering in 1902. During the next four years he was in charge of the rolling-stock department of the Kansai Railway Company, Limited, Osaka, and then spent a year as superintendent of the workshop of the company.

From 1907 to 1926 Mr. Igarashi served as superintendent of the generating station of the Yokohama Electric Company, Limited, at Yokohama, and since that time had been connected with the Ajikawa Iron Works.

Mr. Igarashi also belonged to the Society of Mechanical Engineers of Japan.

CHARLES JABLOW (1887–1935)

Charles Jablow, mechanical engineer in charge of design of Diesel-electric locomotive parts for the Westinghouse Electric & Manufacturing Co., at the South Philadelphia Works, Lester, Pa., died on September 12, 1935, at his home in Drexel Hill, Pa.

Mr. Jablow was born at Louisville, Ky., on December 26, 1887. He attended the Manual Training School at Louisville, from 1902 to 1906, then entered Kentucky State University. He was graduated with a B.M.E. degree in 1909 and three years later received his M.E. degree from the University.

During the summers of 1905 and 1906 Mr. Jablow worked for the Louisville Home Telephone Company and he spent the summers of 1908 and 1909 as draftsman and stockkeeper for the Kentucky Elec-

tric Company at Louisville. In the fall of 1909 he became a designer for the Northern Engineering Works, of Detroit, Mich.

Beginning in 1910 Mr. Jablow spent ten years in the teaching profession. For the first two years he was assistant in machine design and mechanical drawing at the Kansas State Agricultural College and from then until 1920 was located at the Oklahoma Agricultural and Mechanical College, at Stillwater, advancing from assistant professor of mechanical engineering to a full professorship. In 1918 he was also special inspector in the Mid-Continent Field for the Oil Division of the United States Fuel Administration.

In 1920 Mr. Jablow entered the Railway Engineering Department of the Westinghouse Electric & Manufacturing Co. at their East Pittsburgh Works, directing his efforts to the design of the mechanical parts of electric locomotives until 1934, at which time he assumed charge of the mechanical design of Diesel locomotives at the South Philadelphia plant.

Mr. Jablow was the author of many papers pertaining to mechanical design of electric locomotives and was granted many patents covering the detailed parts of such units. His work in connection with the development of the mechanical strain gage as used for the measurement of track stresses was notable. In addition to his professional work, he was associated with many outside activities, such as the Boy Scouts of America, local civic clubs, and local welfare organizations. He became an associate-member of the A.S.M.E. in 1915 and a member in 1920. He also belonged to the Society for the Promotion of Engineering Education, American Association for the Advancement of Science, and the Masonic Order.

Mr. Jablow is survived by his widow, Frances (Train) Jablow, a son, Warren, and a daughter, Alice.

HUGO JUNKERS (1859–1935)

Hugo Junkers was born on February 3, 1859, in Rheydt, near Düsseldorf, where his father owned a weaving establishment.

He attended the Higher Trade School in Barmen, where he thus early showed an unmistakable inclination toward technology and especially machine construction. From 1878 to 1883 he studied machine design at the Polytechnic Institutes at Berlin, Karlsruhe, and Aachen. At Aachen he worked under Slaby, who was then busy with his basic experiments on the four-stroke-cycle engine. When Wilhelm von Oechelhaeuser asked Slaby to recommend a young engineer to cooperate in developing a gas engine at Dessau, he recommended Junkers.

This cooperation between Oechelhaeuser and Junkers brought about the large opposed-piston gas engine, which formed the basis for Junkers' later development of the opposed-piston oil engine.

During this period Junkers developed the gas calorimeter which brought his name before physics students all over the world.

While at Dessau at this period he invented the gas-fired hot-water heater which has been a blessing in countless bathrooms and which is still manufactured in Dessau as well as in many other places in many countries.

Because of the outstanding character of his work at Dessau, Junkers was recalled to Aachen and from 1897 till 1911 was academic instructor in heat theory and director of the newly established machine laboratory at the Polytechnic Institute there.

During these years he carried on many elementary experiments in heat transfer and in structure design, the results of which formed the basis for his subsequent Diesel motor and aircraft work.

In 1911, Junkers resigned from Aachen in order to devote all of his energies to air-transportation problems, which absorbed all of his time until his death, on his 76th birthday (1935), at Gauting, near Munich.

He established a great private laboratory at Dessau. At that time aircraft were built with wooden frames and propelled with heavy gasoline motors. He was the first successful builder of the all-metal airplanes and the first to apply the compression-ignition type of heavy oil engine in commercial air service.

His persistence in thorough fundamental research, beginning while first at Aachen, and continuing throughout his lifetime, combined with the unusual ability to utilize the experimental data for commercial design, made his lifework an outstanding example of useful effort. His great inventions and developments, one after another, were of practical value. The calorimeter has been useful in improving our facilities for technical instruction. The gas water heater added to the cleanliness and comfort in countless homes. The close cooperation between the Research Laboratory at Aachen, and later the Junkers Laboratory at Dessau, and the German manufacturing industry was a pioneer movement which paved the way for the scientists and the industrialists to utilize and profit by each other's ability and experience, as is now being done so generally both in Europe and America. His daring originality foresaw not only the need of the metal airplane but how to make it in a practical way. The result contributed greatly to the speed and safety of modern air transportation.

He realized the limitations of both the carburetor-spark-ignition type gasoline engine and the heavy, slow-speed Diesel motor of the World War days, and recognized that for long distance, high-speed passenger and freight transport more efficient, safer, and lighter weight engines were required. He lived to see his light-weight, multicylinder, high-speed, opposed-piston, compression-ignition engine burning safe, inexpensive fuel oil in actual operation, and since his death it has become commercial.

I remember one day early in 1931 seeing one of his planes with his new aircraft oil engine in it, with Herr Junkers himself aboard, mount gracefully from the air field at Munich. I had inspected the engine the day before at Dessau and seen a duplicate on the test-bed there. There was no doubt that the Diesel-type motor for air service had arrived. It required only five years more for the Diesel to so establish itself as to be used in the transatlantic flights of the *Hindenberg*. While this greatest lighter-than-air ship and its engines are quite different from the planes and motors developed by Junkers, they represent the constructive effect on that branch of German industry of the fundamental, sound, and novel research which he made about thirty years ago.

Industry recognized his work and rewarded him for it. In the scholastic field Professor Junkers was also recognized and honored. The Polytechnic Institute at Munich made him a doctor of engineering; the Department of Philosophy of the University of Giessen gave him a Ph.D.; the Siemens Ring Foundation bestowed the Siemens Ring on him on December 13, 1930, and the V.D.I. gave him the Grashof gold commemorative medal at its National Meeting in 1927. He became a member of the A.S.M.E. in 1924.

His was a life rich in the blessings of successful work, the fruition of much of which he lived to see. He was a simple, modest man who was always ready to recognize merit in his coworkers. He will be long remembered as an outstanding, cultured, German engineer whose enduring work was consecrated to the Fatherland and remains as a monument to his clear vision, his audacity in undertaking new developments, and his persistence in perfecting them.—[Memorial prepared by Oliver F. Allen, New York, N.Y. Mem. A.S.M.E. Based on an appreciation by Dr. A. Nägel, published in *Zeitschrift V.D.I.*, March 9, 1935.

ERNEST RODHAM KENNER (1885–1936)

Ernest Rodham Kenner, manager of tires, accessories, and specialties sales for the Shell Petroleum Corporation, St. Louis, Mo., died on March 23, 1936, of pneumonia.

Mr. Kenner was born at Atlanta, Ga., on September 8, 1885, son of William Hamblen and Lucretia Clay (Tibbs) Kenner. He attended high school at Dalton, Ga., and was graduated from the Georgia School of Technology in 1907, with a B.S. degree in mechanical engineering. Following his graduation he worked for a time as boiler salesman for the R. D. Cole Manufacturing Co. and as draftsman and engineer in the district office of the General Fire Extinguisher Company at Charlotte, N.C. In 1909 he became a student apprentice in the Allis-Chalmers Manufacturing Company at Milwaukee, Wis., and upon the completion of his training course engaged in sales work for that company until 1916. He then took a position as supervising engineer of Plant No. 2 of the Firestone Tire & Rubber Co., at Akron, Ohio. Leaving there in the fall of 1917, he was with the Westinghouse Electric & Manufacturing Co. in Pittsburgh, Pa., engaged in sales and engineering work for the Power Division, until March, 1918. The next year was spent as mechanical superintendent for the Air Nitrates Corporation at Muscle Shoals, Ala.

In April, 1919, he became assistant sales manager for the Wellman-Seaver-Morgan Company, Cleveland, where he remained until the fall of 1923. During the next two and one-half years he engaged in contracting work and from then until the latter part of 1931 he was in charge of automobile tire sales for the B. F. Goodrich Co., Akron. His association with the Shell Petroleum Company dated from the first of January, 1932.

Mr. Kenner married Ethel Lillian Williver, in Milwaukee, in 1913 and is survived by her and by their four children, Rodham W., Forrest, Hamilton, and Elizabeth Kenner.

He became a junior member of the A.S.M.E in 1912 and a member six years later.

JAMES MARTIN KENT (1865–1935)

James Martin Kent, for the past 38 years teacher and engineer for the Kansas City School District, Manual Training High School, Kansas City, Mo., died on March 24, 1935. A son of Richard and

Rosetta (Chambers) Kent, he was born on October 6, 1865, at Kewanee, Ill., and secured his early education in the public schools there and at Wyoming, Ill. He was graduated from the University of Illinois with the degree of bachelor of science in mechanical engineering in 1885. During the remainder of that year he was employed by Cowl & Vandenburgh, Chicago, operating an electric-light plant and working in the shop. Early in 1886 he installed plants for the Sperry Electric Light, Motor & Car Brake Co., in Chicago. He then went to Kansas City, Mo., to install and operate the plant of the Sperry Associated Electric Company. He continued in electrical installation and repair work in Kansas City and vicinity until 1897, during the last nine years serving as chief engineer of the electric light and power plant of the Emery, Bird, Thayer Dry Goods Company, which he had installed.

In 1897 he accepted a position as teacher of steam and electricity and as designer and operator of the power plant in the Manual Training High School. In addition to this work he carried on a small practice as a consulting electrical and mechanical engineer, and during the period 1912–1917 was president of the Henrici, Kent & Lowry Engineering Co., during which time he was largely responsible for the design and installation of a number of municipal and private plants.

He served for some years as a member of the Board of Examining Engineers of Kansas City, and was engineer for the Board of Education. During the World War he was engineer for the United States Fuel Administration in the Kansas City District.

Mr. Kent became a member of the A.S.M.E. in 1900, and in the same year joined the American Institute of Electrical Engineers. He was also a member of the American Society of Heating and Ventilating Engineers, Institute of Radio Engineers, American Chemical Society, American Electrochemical Society, American Association for the Advancement of Science, American Forestry Association, Society for the Promotion of Engineering Education, National Society for the Promotion of Industrial Education, National Educational Society, Missouri Society of Mathematics Teachers, National Association of Power Engineers, National Geographic Society, and Kansas City Engineers Club.

Surviving Mr. Kent are his widow, Jessie Paull (Nichols) Kent, whom he married in 1893, and one son, Paul N. Kent.

DAVID J. KERR (1874-1934)

David J. Kerr, general operating superintendent of The Champion Fibre Company, Canton, N.C., died in his berth on the night of June 19, 1934, while traveling from Canton to Raleigh, N.C., to attend the North Carolina State Convention.

Mr. Kerr was born at Ballybay, County Monaghan, Ireland, on July 19, 1874. His father died in 1885 and the mother and family moved to America in 1887, locating in Atlanta, Ga. David attended the West End Academy there until after the death of his mother in 1890, being forced to leave school at that time in order to help support the younger members of the family. At that time the first electric street railway was being built in Atlanta and for two years Mr. Kerr was in charge of the motor equipment. In 1893 he secured work as a wireman in connection with preparations for the Cotton States and International Exposition to be held in Atlanta in 1895. When the Exposition opened he was appointed to the position of electrical inspector and at its close was given a place as assistant superintendent of construction for the Tennessee Centennial Exposition which was to be held in Nashville in 1897. During the operation of this Exposition he was superintendent of maintenance of the electrical department. At the close of the Centennial Mr. Kerr moved to Rome, Ga., where for five years he was superintendent and chief engineer of the Rome Railway & Power Co. He resigned this position to become electrical engineer for the Tennessee Copper Company, at Copper Hill, where he later served as master mechanic in addition to other duties. Correspondence courses in engineering in 1904–1905 supplemented his early schooling.

Mr. Kerr entered the employ of The Champion Fibre Company as chief electrician in 1908. In 1915 he built and equipped an electrolytic bleach plant and in 1921 he was appointed superintendent of power, operating the three departments, steam, electrical, and electrolytic bleach. He was appointed general operating superintendent on October 1, 1925.

Mr. Kerr was a genial, friendly type of man who, though recognized as having peculiar qualifications for leadership, was known to almost every one as "Dave" or "D.J." In his new position he still answered to his nickname.

During the World War Mr. Kerr made exceptional records in putting Canton many times "over the top" in Red Cross and Liberty Loan Drives. He was elected mayor of Canton in 1923 and reelected each term until the year of his death. Canton owes her well-paved streets to him. He took an active part in state and national politics and was well known from one end of the state to the other. He was also an enthusiastic worker in the church and Sunday School of The Southern Methodist Church and in the Champion Y.M.C.A., as well as in several fraternal orders, most especially the Knights of Pythias and Dramatic Order Knights of Khorrassan. He held the station of Chancellor Commander in Canton Lodge No. 149 for three terms, beginning in January, 1909, and was District Deputy Grand Chancellor for several years, beginning in 1910. He joined Suez Temple No. 73 of the Dramatic Order Knights of Khorrassan in 1910 in Asheville, N.C., and was largely responsible in the organization of Bagdad Temple No. 213 of the Dramatic Order Knights of Khorrassan in Asheville, of which he was a Past Royal Prince in 1918.

Mr. Kerr had been an associate member of the A.S.M.E. since 1920. He served as vice-chairman in 1924 and chairman in 1925 of the Greenville Branch of the Carolinas Section of the Society. He also belonged to the American Institute of Electrical Engineers. He became a naturalized citizen of the United States when he reached the age of twenty-one.

Mr. Kerr was married in 1893, his wife being Miss Kate Joiner, a daughter of Captain and Mrs. John J. Joiner, of Atlanta, Ga. He is survived by her and by three children, Mrs. Marie (Edw. W.) Bell, of Canton, N.C., Mrs. Bertram Cedric Hope, of Grand Rapids, Mich., and D. Seymore Kerr, mechanical and electrical engineer with the Allis-Chalmers Company, Chattanooga, Tenn.

DWIGHT FOSTER KILGOUR (1863-1935)

Dwight Foster Kilgour, whose death occurred at Lexington, Mass., on September 10, 1935, was born at Lynn, Mass., on January 5, 1863, son of Hannibal Colby and Mary Alice Kilgour.

Leaving school when he was fourteen, Mr. Kilgour was employed as an errand boy by a department store in Boston for about two years. During this time he picked up his first knowledge of engineering by helping the engineer and fireman in the adjoining building during his spare time. He next worked for the Waters Governor Works for a few months, running small machines, then worked at several temporary firing jobs. For several months, when he was seventeen, he was night fireman at the Jordan Marsh & Co. department store and for the next two winters was night fireman for the Walworth Manufacturing Company in a large block plant. During the intervening summer he ran a large steam engine and fired the boiler for the Commercial Manufacturing Company (Oleomargarine Manufacturing Company). He also fired on towboats for the Boston Tow Boat Company, and in the spring of 1882 took a job as oiler on the excursion steamer *Rose Standish* of the Nantasket Steam Boat Company. On his first trip, quick action by him, when the engineer had temporarily left the engine room, prevented a serious collision with a ferry boat. This incident led to his appointment as second engineer.

These positions gave Mr. Kilgour good experience with different types of engines and boilers—experience on which he based important contributions to steam generation in later years.

In the fall of 1882 Mr. Kilgour went to work for Charles E. Cotting, a young man only four years his senior, who was just starting in the real estate business for himself in his father's office in Boston. The association continued for nearly forty years, until the death of Mr. Cotting in 1921.

In a review of his work for Mr. Cotting prepared by Mr. Kilgour about a year before his death he says:

"Mr. Cotting started me in as an engineer in the Old Boston Music Hall. There was one elevator engine, one engine for pumping wind for the large organ, and three horizontal-return tubular boilers. In the year of 1885 or 1886, incandescent electric lighting was installed throughout the Music Hall, also two units of generating apparatus—that being only the second isolated plant in Boston at that time."

As Mr. Cotting acquired more property, he made Mr. Kilgour his superintendent and engineering adviser. The first large project undertaken by them was the building of the State Street Exchange (Stock Exchange Building), the largest office building in New England at that time. Mr. Kilgour wrote further:

"I took superintending charge of that building in 1893, with my mind continually on future projects. In the year of 1895, we built the Tremont Building, Parker House Extension, and the Hotel Touraine—and for quite a few years we always had two and sometimes three new buildings in process of construction—such as Symphony Hall, Henry Segal Department Store, South Terminal Trust Buildings, South Street Trust Buildings, Kimball Building, Post Office Square Building, Paddock Building, Gilchrist Building, and many others, including bank buildings (those with which Mr. Cotting was connected as director), theatres, manufacturing buildings, etc. In the year of 1900, Mr. Cotting took over the Boston Real Estate Trust Buildings, consisting of sixty large pieces of property. In 1905, at

the death of his father, Mr. Cotting took over his father's business, consisting of the Fifty Associates Properties, Sears Real Estate Trust, Snow Associates Trust, etc. When Mr. Cotting died in 1921, he had 222 pieces of real estate to manage as active trustee, etc."

Of these buildings, the most of which were built under the direction of Mr. Kilgour and his assistants, 75 per cent were operated under his supervision. He indicated the responsibility of his position when he wrote:

"We had over eighty power and heating plants of all descriptions—four block plants for the sale of power, light and heat—over six hundred elevators of different types—over six hundred employees. We were the third largest user of coal in Boston, using nearly eighty thousand tons per year."

After the death of Mr. Cotting in 1921 Mr. Kilgour went into business for himself as consulting engineer, specializing in appraisal work. He made appraisals of various plants, including those of the Walter Baker Chocolate Company at Dorchester, Mass., and Montreal, Canada. In 1923, this company engaged him to install hydrolectric plants at two dams on the Neponset River at Dorchester. One of these was the first multiple automatic plant in the United States. He was retained by the company in general plant improvement work until it was absorbed by the General Foods Corporation in 1927. One of the improvements which he brought about was the installation of a centralized fire alarm station which reduced the fire insurance rates from 33 to 18 cents.

During the remaining years of his life Mr. Kilgour engaged in appraisal and general engineering work for large industrial plants and devoted considerable time to new developments in boilers, furnaces, and mechanical devices.

Of a number of patents held by Mr. Kilgour he considered two of basic importance in development of modern methods of steam generation. In 1909 he patented a secondary combustion tube or "throated tube," to be used in connection with a steam-boiler furnace setting, which was later recognized as the stepping stone to subsequent work in turbulence.

The second invention on which Mr. Kilgour placed particular value was his water-tube circulating system for the protection of wide walls, arches, etc., of steam-boiler furnaces, with headers on the outside of the setting. This was patented in 1919 and although Mr. Kilgour was not in a position to push its development, he felt that it was the pioneer step in water-wall construction.

About 1920 he devised a system for removing the humidity from the atmosphere, for the Nestor Cigarette Company, Boston, which he realized later was pioneer work in air conditioning.

About this time he also eliminated smoke nuisances at the South Terminal Trust power plant by springing an arch across the fireboxes of the locomotive-type boilers. He was asked if he intended to patent this "arch" construction, but did not think it of much importance then.

In 1925, while still with the Walter Baker Company, he devised a new form of brine cooler for refrigeration, using smaller tubing with less surface than had previously been employed, and thereby obtaining thirteen degrees temperature drop, as compared with nine degrees, and increasing capacity nearly fifty per cent.

In 1933 he patented a new type of steam boiler—a vertical firetube boiler with a water-tube circulation which gave excellent results in tests made at the Massachusetts Institute of Technology.

Other patents by Mr. Kilgour included several forms of lubricating devices, hydraulic shock absorbers for automobiles, filter boxes for boiler feedwater, and sprinkler alarm valves.

Mr. Kilgour became an associate of the A.S.M.E. in 1905 and a member four years later. He also belonged to the Masonic fraternity.

Mr. Kilgour was twice married, first to Grace A. Barrett, of Boston, and in 1901 to Rhoda A. Furbish, of Presque Isle, Me. He is survived by his widow and by three children of his first marriage, Alice Louise, Edith Cogswell, and Walter Malcolm Scott Kilgour.

ARTHUR HENRY KNEEN (1872–1935)

Arthur Henry Kneen, whose death occurred at his home in Scranton, Pa., on July 7, 1935, was born at Derby, Conn., on December 27, 1872. His parents were Thomas and Mary (Cheshire) Kneen.

Following his graduation in 1891 from the Michigan Agricultural College with a B.S. degree in mechanical engineering, Mr. Kneen took charge of the Mechanical Department of the New York Inter-Urban Water Company, Mt. Vernon, N.Y., where he remained until 1905. He next spent a period of even greater length with the Operating Department of the American Pipe & Construction Co., Philadelphia. He was engineer and assistant general superintendent until 1917, when he was promoted to the position of general superintendent. He had charge of the design, installation, and operation of all the mechanical equipment of the utilities operated by the American Pipe & Construction Co., which consisted of water companies, gas companies, electric light and street railway companies, and sewerage companies, located in Pennsylvania, New Jersey, New York, Virginia, Georgia, Alabama, Arkansas, and Texas. He designed and installed in many of the plants complete stations consisting of boilers, pumps and steam turbines, piping and full equipment. Subsequently he became chief engineer for the Phila-Suburban Water Company, Philadelphia, designing and building filter beds, sewage plants, and reservoirs. He planned a new large storage lake reservoir, and built a dam at Crum Creek, a new pumping station at Pickering Creek, installing pumps and interior filtration beds, and also a dam to control flow and an aeration system. Since 1927 he had been managing superintendent, operating vice-president, and engineer for the Scranton-Spring Brook Water Service Company at Scranton, and president of subsidiary companies in Pennsylvania.

Mr. Kneen had been a member of the A.S.M.E. since 1919 and was active in water-works associations. He served as vice-president of the Pennsylvania Water Works Association from 1929 to 1934 and was a member of the American Water Works Association, and the Pennsylvania Water Works Operators Association. He was a member of Irem Temple, A.A.O.N.M.S., the Irem Country Club, at Dallas, the Scranton City Club, the Green Ridge Club, and the Engineers Club of Philadelphia. He served as president of the Borough Council of Lansdowne, Pa., from 1922 to 1926.

Married in 1897 to Miss Alice M. Beaudry, he is survived by her and by their son, Thomas Beaudry Kneen.

ARTHUR E. KRAUSE (1853–1935)

Arthur E. Krause, who died on July 1, 1935, at Mountain Lakes, N.J., was a native of Germany. He was born at Zwickau, Saxony, on April 2, 1853, son of Herman Frederick and Sara (Michaelis) Krause. The most of his life was spent in this country. He attended public school in New York, N.Y., and took a course in architectural and mechanical drawing at Cooper Institute. At the early age of seventeen he became a mechanical draftsman for the Department of Public Parks in the City of New York. Under the direction of Jacob Wrey Mould, architect in chief, he made detailed structural drawings for the public buildings in Central Park.

For about two years, beginning in 1880, Mr. Krause served as a draftsman in several architects' offices in New York, and in 1882 became mechanical draftsman on sugar machinery for the F. O. Matthiessen & Wiechers Sugar Refining Co., Jersey City, N.J. He was made structural and mechanical engineer for that company the following year, and retained that position until 1887. From then until 1908 he was structural engineer for the American Sugar Refining Company, Jersey City, and during the remainder of his active life engaged in the manufacture of filters of various kinds. He held patents on methods of clarifying and re-boiling water, particularly for ice manufacture, on a re-boiler and condensate purifier, and on other filtering equipment. He contributed articles to the technical press on sugar machinery and on the purification of steam condensate.

Mr. Krause had been a member of the A.S.M.E. since 1887 and belonged to the Mountain Lakes Men's Club. He was unmarried and is survived by a sister, Miss Clara Krause, of Mountain Lakes.

ROSCOE RAYMOND LEONARD (1894–1935)

The sudden death on December 8, 1935, of Roscoe Raymond Leonard, since 1926 a member of the secretarial staff of the A.S.M.E. and since 1929 representative of the Society at its Mid-West office in Chicago, was reported in *Mechanical Engineering* as follows:

"Mr. Leonard had come to New York for the annual meeting of the A.S.M.E. and when last seen by his associates on Saturday, December 7, was apparently in the best of health and spirits. Death, several hours later, resulted from a cerebral hemorrhage and was almost instantaneous.

"Mr. Leonard became a member of the A.S.M.E. headquarters staff in 1926 and for several years served in the editorial department. He had previously been associate editor and assistant secretary of the American Society of Refrigerating Engineers. His advent to the A.S.M.E. staff was at a time when the Society's publications were undergoing drastic revision and enlargement as a result of the change from an annual volume of Transactions to a series of publications comprising the papers presented at the meetings of the Professional Divisions. To the extensive correspondence necessary to make clear the fact and purposes of this change to members who were confused by it, he devoted much time. Members of the Fuels Division will recall his services in connection with the widely attended national meetings of that division which marked the unusual technological develop-

ments in stoker and pulverized-fuel practice of the late twenties. He also edited a question-and-answer department under the auspices of the divisions that was a feature of *Mechanical Engineering* for a few years.

"When the A.S.M.E. Council voted, in 1929, to open a Mid-West office in Chicago, in order to bring the services of the Society's headquarters staff closer to members in that district, Mr. Leonard was assigned to the office. A native of Indiana and a former student at Purdue University, he brought to his duties in that post a sympathetic understanding of its problems. Among the members in Chicago and vicinity he made many warm friends. His visits to the Local Sections and Student Branches in the territory served by his office were of value to the Society and to the members in that district who felt that they had in him a personal representative. His two hundredweight of geniality won many friends, and his willingness to be of service will be sorely missed."

Mr. Leonard was born at Pekin, Ind., on April 23, 1894, son of William and Bertha (Tash) Leonard. The family moved to Salem, Ind., when he was thirteen years old and he was graduated from the high school there in 1913. He studied electrical engineering at Purdue University the following year, but becoming interested in voice culture, entered the Indianapolis Conservatory of Music, from which he was graduated in 1915. He studied a short time, also, in the Louisville Conservatory of Music. He married Miss Vera Albin, a teacher at the Indianapolis conservatory, in 1916.

After some early experience as a draftsman for companies in Indianapolis and Louisville, while studying music, Mr. Leonard went to Cincinnati, in 1916, to take a position in the engineering department of the Triumph Ice Machine Company. He was there until 1919, and during the next year was connected with the ice-machine department of the De La Vergne Machine Company, New York. His work for these two companies consisted of drafting, which entailed refrigeration computations, plans for machinery and piping installations, and design of ice-manufacturing equipment. For a short time in 1920, prior to becoming assistant secretary to The American Society of Refrigerating Engineers, he worked with T. Howard Barnes, consulting engineer, New York, on engineering and drafting in connection with a large packing house for Colombia, S.A.

An editorial in *Refrigerating Engineering*, February, 1936, read, in part, as follows:

"Mr. Leonard was attached to the staff of The American Society of Refrigerating Engineers from 1920 to 1926, serving as editor of *Refrigerating Engineering* from its inception in 1922. He was a junior member of the society during the first of the period, and later was promoted to full membership, which he retained until 1933. He was secretary of the Refrigeration Safety Code Committee for a period of seven years, during his employment by the A.S.R.E. and by the A.S.M.E., and held other committee jobs in both groups.

"As we knew him, those virtues were his which we all look for in men and seldom find—fundamental honesty and loyalty, frankness of purpose and action. He bore malice to none, was remarkably accommodating and good natured, but not afraid of a fight when one was necessary. With rather meagre advantages at the start he rose to a place of influence in his profession, an influence that will be felt after him."

Mr. Leonard was markedly active in music and church work. He was a soloist at the First Methodist Episcopal Church in Montclair, N.J., organized the Clarion Quartet, which filled many engagements in the Metropolitan area, and sang occasionally over radio station WJZ while he was in the East. After moving to Chicago he joined the Norwood Park Methodist Episcopal Church and was a member of its official board, chairman of the music committee, and director and bass of the church quartet. He took great pleasure in the cultivation of flowers in the gardens surrounding his home, and was devoted to the care of his wife, for a number of years an invalid. He is survived by her and by two children, Betty Jane and Robert Albin Leonard, as well as by his parents and a brother, Jesse Leonard, of Clarksburg, W.Va.

DAVID J. LEWIS, JR. (1857–1935)

David J. Lewis, Jr., consulting engineer, a member of the A.S.M.E. since 1892, died on December 26, 1935.

The son of David J. Lewis, a marine engineer, and Anna E. Lewis, he was born at Baltimore, Md., on Christmas Day, 1857. His education, through high school, was secured in the public schools of Chester, Pa., and he was graduated from the Polytechnic College, at Philadelphia, as a mechanical engineer. After serving an apprenticeship in the boiler and machine shops and drafting room of J. Roach & Son, shipbuilders, at Chester, he spent some time at sea, reaching the grade of assistant engineer on ocean steamers.

The Polytechnic College gave him a master's degree in mechanical engineering in 1881, and during the next two years he was superintendent of the Dobbins Boiler Works at Lowell, Mass. He then returned to the Roach shipyard where he was assistant foreman of the Boiler Department until 1886.

During the next four years Mr. Lewis was superintendent of the Boiler Department of the Southwark Foundry & Machine Co., in Philadelphia. He next went to Chicago, where he spent two years as general superintendent and engineer of the Porter Boiler Manufacturing Company, Standard Fireless Engine Company, and the Boiler Department of the Excelsior Iron Works.

In 1892, after a short time as engineer in the Standard Pipe Department of the Stearns Manufacturing Company, Erie, Pa., Mr. Lewis became superintendent of the Gill Boiler Works, Philadelphia. He then returned to Lowell where he held the position of chief engineer of motive power of the Merrimac Manufacturing Company.

From 1897 on Mr. Lewis was associated successively with a number of companies as consulting engineer and sales manager. The first of these was the A. A. Griffing Iron Co., of New York and Boston, which he served until 1909. Then there came two three-year periods with the American Radiator Company and Lyten Manufacturing Company, both of New York. From 1915 to 1920 he was connected with the W. J. Wayte Co., New York, and during the next ten years with Cresson & Morris, Philadelphia. In December, 1930, he became consulting engineer and sales agent of the Centrifugal Department of the Fletcher Works, with headquarters in New York, and he continued in that position until his death.

Mr. Lewis married Miss Leila F. Kinder, of Bridgeville, Del., in 1882. Her death occurred on January 1, 1936, less than a week after his. They are survived by two children, Walter Harold Lewis and Ethel (Lewis) Hirsh.

JOHN J. LICHTER (1867–1935)

John J. Lichter, consulting engineer, St. Louis, Mo., a member of the A.S.M.E. since 1924, died of heart failure at Estes Park, Colo., on June 13, 1935. He was a native of St. Louis, having been born there on May 27, 1867, son of John and Mary Emma (Arnoux) Lichter. His education was begun in St. Louis and continued in Denver, where his family moved because of the ill health of his mother. After her death in 1883, he returned to St. Louis and entered the St. Louis Manual Training School, from which he was graduated two years later. He then entered Washington University, where he spent five years in the study of mechanical and electrical engineering, graduating in 1890.

Shortly after his graduation he entered the employ of the late John Scullin, who became one of his best friends. In addition to his engineering work for Mr. Scullin, he taught astronomy at Washington University from 1897 to 1899, and again from 1901 to 1903.

From 1894 to 1898 he was engaged in electric-railway work in charge of all engineering for the Union Depot Railroad (electric) Company, of St. Louis, until its consolidation with the United Railways Company, when he became chief electrical engineer of the United Railway system.

In 1901 he became associated in consulting engineering with William Jens, a civil engineer, who had done valuable work in laying out railroads in Mexico. The firm received commissions for the design and construction of buildings and equipment for power stations for many public-utility companies in Missouri and Nebraska, including the East St. Louis & Suburban Railway, Omaha & Council Bluffs Street Railway, East St. Louis, Columbia & Waterloo Railway, St. Joseph Railway, Light, Heat & Power Co., and Lincoln Gas & Electric Co.

They also designed buildings and power equipment for industrial firms, including the Heine Boiler Company, St. Louis Frog & Switch Co., Fulton Iron Works, Scullin Steel Company, St. Louis Basket & Box Co., Valier & Spies Milling Co., Geo. P. Plant Milling Co., Evertight Piston Ring Company, Warner Chemical Company, Cupples Company, and Fulton Bag & Cotton Mills, all in St. Louis; also a heating plant for the Missouri Botanical Garden, buildings for the Dennison (Tex.) Light & Power Co., the Ralston Purina Company, at Fort Worth, Tex., a power station for the Walsh Fire Clay Products Company, a tVandalia, Mo., and other plants, the total expenditures for which ran into millions of dollars.

The St. Louis Frog & Switch Co. was organized by Mr. Lichter in 1906 and he served as its president until its liquidation in 1932.

The Bulletin of the Academy of Science of St. Louis for November, 1935, contains resolutions adopted by the Council of the Academy from which the following is taken:

"One seldom meets a man of such broad interests and versatility of mind as were possessed by Mr. Lichter. He had an almost profound knowledge of many subjects, such as electrical and mechanical engineering; of astronomy, physics, and geology. He was also well ac-

quainted with the fauna and flora of many parts of our country. He was a collector of butterflies and raised fine chickens, horses, and cattle as a side issue, often receiving prizes at exhibitions.

"A photographer since his boyhood, he recently became interested in taking motion pictures of wild life both in black and white and in color. He kept up with the advances in many branches of science.

"Mr. Lichter was also a business man of unusual ability and integrity, wonderfully well-informed concerning investment securities, especially when investing the funds of others.

"He was also very fond of music. When a student at Washington University, he organized and was leader of the orchestra. He has been a member of a string quartet for 28 years. He played the violin and was the possessor of several good instruments. This quartet met at Mr. Lichter's home on Monday evening of each week until within a few weeks of his death.

"As a result of his keen business ability, he naturally accumulated a small fortune and he was most generous with this. On the death of his wife, whom he greatly mourned, he built, as a memorial to her, the Humane Society Animal Shelter.

"Mr. Lichter was a life member of the Academy of Science of St. Louis and very active on its Council. He was a member of the St. Louis Engineers' Club, the St. Louis Institute of Consulting Engineers, the Circle Club, the American Association for the Advancement of Science, the Missouri Academy of Science, and a life member of the American Museum of Natural History."

Mr. Lichter's wife, Irene V. (Anderson) Lichter, whom he married in 1902, died in 1928.

ARTHUR DEHON LITTLE (1863–1935)

Arthur Dehon Little was born on December 15, 1863, at Boston, Mass., attended the public schools at Portland, Maine, and the Berkeley School in New York, and married Henrietta Rogers Anthony of Boston on January 22, 1901. He died on August 1, 1935.

While an undergraduate at the Massachusetts Institute of Technology he participated in founding the student newspaper on which he served as editor in chief. After graduating in 1885, Doctor Little kept in active touch with the Institute throughout his life. In helping to initiate the School of Chemical Engineering Practice he made an outstanding contribution to scientific education. He served on visiting committees at both Harvard and M.I.T., and was of constructive assistance to many other educational institutions, particularly in recommendations for strengthening the curricula of courses in chemical engineering. He participated in the founding of the *Technology Review*, and served as president of the Alumni Association in 1921–1922 and as its representative on the Corporation from 1912 until his election as a life member in 1922.

Upon graduating in 1885 he became superintendent of the first American sulphite-pulp mill. In 1886 his imagination and courage enabled him to establish what was probably the earliest private consulting organization in America devoted to the application of science to industry. His partner, R. B. Griffin, was killed in an accident in 1892, and for seven years thereafter Doctor Little carried on the business alone, forming a new partnership with Dr. Wm. H. Walker in 1900 which he continued until 1909 when the business was incorporated. In 1917 his laboratories were removed to their present location in Cambridge.

In the early days of Doctor Little's organization there was no appreciation in this country of the place of science in industry and for many years he was a pioneer. The fact that he was able to break down the barriers of ignorance, secrecy, and prejudice and give a new impetus and direction to the industrial development of the country speaks eloquently for Doctor Little's power to inspire, to persuade, and to lead.

He had an almost uncanny instinct for fertile, unexploited territory, and directed and stimulated research which resulted in new industries and the development of important world commodities. Doctor Little took out patents and processes for the manufacture of chrome-tanned leather, chlorate of potash, and cellulose acetate, and later invented processes for making newsprint from southern woods and for the recovery of naval stores from lumbering wastes. During the War he acted as consultant to both the Chemical Warfare Service and the Signal Corps. He was in charge of specialre searches on airplane dope and acetone production, and was the inventor of a smoke filter which became part of the standard equipment of the United States Army.

In a tribute read at a meeting of the Corporation of the Massachusetts Institute of Technology, in which he was deeply interested, it was said that Doctor Little's "greatness rests not so much upon specific inventions as upon the many advances in many fields for which he was responsible through inspiration and leadership. Always generous and open in his disclosures of information, he did much to break down the narrow and hampering secrecy that characterized American industry before his day.

"Transcending his accomplishments in chemical engineering, in the administration of scientific enterprises, and in education, stood the man himself, distinguished and gracious in manner, gifted in speech, wise and unselfish in counsel. He was a companion sought after for his charm, his sincerity, his urbane humor, and his learning. His scholarship, his vital interest in and knowledge of the work of his contemporaries, his delightful personality, and his gifts as a speaker and writer made him a most effective interpreter of the ideals and objectives of science, as applied to industry. Collateral with his capacity to appreciate and to produce a high type of interpretative literature was a highly developed esthetic sense which found its most specific expression in his rare and beautiful collection of Chinese porcelains."

His international eminence was indicated by the many honors bestowed upon him, including his elections to the presidencies of the American Chemical Society, the American Institute of Chemical Engineers, and the Society of Chemical Industry in London. He was awarded the Perkin Medal in 1931 and received the highest honorary degrees from the University of Pittsburgh, Columbia University, Tufts College, and, in England, the College of Technology at Manchester and the University of Manchester.

The Columbia citation as drafted by Nicholas Murray Butler is especially fitting and gratifying to his multitude of friends: "Arthur Dehon Little, Chemical Engineer—Native of Massachusetts; a captain in the organization and direction of research in the science of chemistry in all its manifold revelations covering in his field of interest and influence almost every aspect of chemical-engineering practice; fertile in invention, practical in application and a genuine leader in the preservation and advancement of that organized body of knowledge which we know as science; one who, as even Sir Humphry Davy would admit, pursues science with true dignity."

Doctor Little was a member of American Academy of Arts and Sciences, American Association for the Advancement of Science, American Chemical Society, American Electrochemical Society, American Gas Association, American Institute of Chemical Engineers, American Institute of Mining and Metallurgical Engineers, American Petroleum Institute, The American Society of Mechanical Engineers (since 1912), Chemical Society (London), The Franklin Institute, Engineering Institute of Canada, The Institute of Fuel (London), Institution of Petroleum Technologists (London), Royal Society of Arts (London), Society of Chemical Industry (London), Société de Chimie Industrielle (New York Section).

He published many articles, notable not only for their scientific and humane value, but for the clear and individual style in which they were written. Among his best known literary works are "Chemistry of Paper Making" (with R. B. Griffin), 1894, "The Fifth Estate," 1924, and "The Handwriting on the Wall," 1928. The book on paper making was for three decades the one authoritative technical reference work in the industry.—[Memorial prepared by Prof. W. H. McAdams, with the cooperation of Dr. Vannevar Bush, John J. Rowlands, Dr. F. G. Keyes, and Raymond Stevens.]

CHESTER BRADFORD LORD (1870–1934)

Chester Bradford Lord, for five years associate editor of *American Machinist*, died on May 26, 1934, as the result of an accident which occurred near Sellersville while he was visiting in Pennsylvania.

Mr. Lord was born at North Milford, Me., on December 9, 1870, the son of Charles and Nancy Lord. After leaving high school he began work in 1887 with the New York Central & Hudson River R.R., becoming shop foreman in 1890 and continuing in that position until 1894. In 1896 he took the position of assistant superintendent of the Tubular Dispatch Company, and was engaged until 1898 in the installation and operation of pneumatic mail tubes in New York. He then went to Chicago where he served the Chicago Great Western Railroad as shop foreman for two years.

In March, 1900, Mr. Lord became machinist for the General Electric Company, Schenectady. He was made foreman of the Train Control Department in the fall of 1903 and held that position until October, 1905, when he went to St. Louis, Mo., to superintend the construction, layout, and equipment of a new plant for the Wagner Electric Manufacturing Company. He remained with this company until about 1920, serving as general superintendent and consulting engineer.

The next position held by Mr. Lord was that of manager for the Advance-Rumely Company, a farm implement firm, of Battle Creek, Mich., which was later absorbed by the Allis-Chalmers Manufacturing Company. About the middle of the year 1922 he became general superintendent for The National Automatic Tool Company, Richmond, Ind., with which he remained until August, 1923. He is next

recorded, in 1924, as a member of a Chicago firm of consulting engineers, Wallace, Delany, and Lord, and the following year became superintendent of shops for the Newport News Shipbuilding & Dry Dock Co., Newport News, Va. He remained there until he became a member of the staff of *American Machinist* in 1929.

Mr. Lord joined the A.S.M.E. in 1913, served on the regular Nominating Committee in 1920, and contributed a number of papers to the meetings of the Society. He was at one time a member of the group committee for the Dominican Republic of the Inter-American High Commission, which was established at the Pan American Financial Conference of 1915 and had for its purpose the development of uniform legislation in the republics of the American Continent. Mr. Lord also had some connection with the perfection of bomb discharges at Mare Island during the World War, and served on the Board of Parks and Playgrounds in St. Louis.

Mr. Lord married Miss M. May Stimson in 1901 and is survived by her and by two children, Frances and Roger Lord.

MARCUS THOMPSON LOTHROP (1884–1935)

Marcus Thompson Lothrop, a consultant in metallurgy and anti-friction bearings and a pioneer in the manufacture of tapered roller bearings, died at his home, Hills and Dales, Canton, Ohio, on May 24, 1935. He was former president of the Timken Roller Bearing Company and of its subsidiary, the Timken Steel & Tube Co., of Canton.

Mr. Lothrop was born at Buffalo, N.Y., on October 2, 1884, son of Dr. Benjamin L. Lothrop and Maria (Thompson) Lothrop. After graduation from the Buffalo High School he entered the College of Literature, Science and Art of the University of Michigan. He received the degree of bachelor of science in chemical engineering in 1906 and in 1932 the University conferred upon him the honorary degree of master of engineering.

For a year following his graduation in 1906 Mr. Lothrop was assistant to Professor White at the University in work on iron and steel. On April 1, 1907, he took charge of the physical laboratory of the H. H. Franklin Manufacturing Co., Syracuse, N.Y., working on materials for automobile construction. After a short time there he took the position of metallurgist for the Halcomb Steel Company, Syracuse, with which he remained until becoming associated with the Timken Roller Bearing Company on October 1, 1912, as manager of the steel and tube department. He had taken out various metallurgical and anti-friction bearing patents.

Mr. Lothrop became a junior member of the A.S.M.E. in 1908 and a member in 1917. He served on the Subcommittee (of the Committee on Meetings) on Iron and Steel from 1912 to 1915. He was also a member of the American Institute of Mining and Metallurgical Engineers, American Society for Testing Materials, American Society for Steel Treating, American Iron and Steel Institute, Society of Automotive Engineers, and the British Iron and Steel Institute. His clubs included the Brookside Country Club, Canton; Tam-O-Shanter Country Club, Detroit; Congress Lake Country Club, Hartville, Ohio; Union Club, Cleveland; Detroit Athletic Club; and The Recess, Detroit. He was a trustee of the Aultman Hospital, Canton.

Surviving Mr. Lothrop are his widow, Margaret (Frank) Lothrop, whom he married in 1922, and two children, John Henry and Margaret Lothrop.

FRED ROLLINS LOW (1860–1936)

Fred Rollins Low, past-president of the A.S.M.E., died on January 22, 1936, at his home in Passaic, N.J., after a lingering illness of several years.

Engineering annals are replete with the names of those who have contributed to its advancement, but for the greater part their achievements have been those of the specialist. Mr. Low's role was different—in a sense unique—but not less important. The forty-year period which spanned his active participation in engineering affairs witnessed the most spectacular advance in power-plant practice and the momentous influence of electricity on the industrial growth and social life of the nation. His part in the advancement of this phase of the engineering profession was threefold, namely, his leadership as an editor, his service to the A.S.M.E. and other engineering societies, and his influence at large in the field of power engineering, through his many contacts and wide acquaintance.

As an editor it was Mr. Low's privilege to gather, record, and disseminate information, to interpret advances in practice, and to guide opinion. In this he was progressive, yet cautious to the extent of always making sure of his facts. Ever tolerant with those who differed with him, he accorded them opportunity to express their views fully, either verbally or in print, but having arrived at a decision as to what was in the best interests of the profession, he tenaciously and fearlessly pursued his course regardless of adverse influences. He delighted in exposing proprietary nostrums and panaceas, and, in a larger sense, insisted on telling the facts concerning failures of equipment to the end that safer and better designs might be demanded and evolved. His thinking in such matters was always constructive. In this connection many will recall his campaign in steam-boiler design against "lap-joint" construction and the early practice of designing dished heads with a small corner radius and installing them so that they could not be thoroughly inspected; also his attitude concerning a prevalence of turbine-disk failures that occurred in an earlier period of turbine development. He was an active exponent of mandatory boiler inspection and licensing of operating engineers and had a leading part in putting through such legislation.

Mr. Low always regarded the editorial function as analogous to that of teaching, and his facility for direct analysis and simple expression exerted a widespread educational influence in the field, particularly among operating men. This educational work was not confined to writing but also extended to the platform and he devoted much time, especially in the earlier years, to lectures before operating groups. Aside from this his versatility and breadth of vision encompassed the whole field of power engineering, and as one commentator has aptly expressed it, "He saw eye to eye with those industrial leaders who were concerned with the larger aspects of power production and use."

To his associates in editorial work he was ever an inspiration. While exacting as to the quality of their work and insistent upon strict adherence to prescribed editorial standards, he accomplished this objective through kindly suggestion and by setting the example. Never jealous of his own prerogatives or personal prestige, he encouraged and assisted those about him to develop to the maximum of their capabilities and he never failed to give full credit for work well done. His personal characteristics and human qualities endeared him to all.

Fred Low joined The American Society of Mechanical Engineers as an associate in 1886 and became a full member in 1912. A firm believer in the axiom, "that the value of membership in an engineering society is in direct proportion to what the individual puts into its activities," he early took an active part in its work, serving successively upon many important committees, serving as a vice-president in 1918–1920, and finally being elected as its president in 1924. His two most outstanding contributions to the Society were in connection with the formation and functioning of the Power Test Codes and the Boiler Code, serving as chairman of committees on both until the time of his death. The chairmanship of these two committees required not only technical judgment and foresight but extreme tact and at times firmness in dealing with conflicting opinions and personalities. This was especially so in the work of the Boiler Code Committee where a single adverse vote in the final ballot on one of the interpretations or proposed revisions resulted in referring it back to the committee. His success in this work is attested by the accomplishment and prestige of these codes.

Other activities of the Society in which Mr. Low took an active part were:

Formation of the Gas Power Section (the first Professional Division of the Society, organized in 1908) and its chairman in 1909.
Gas Power Executive Committee, 1910–1914
Fuels Division, Executive Committee, 1920–1923
Professional Divisions, Standing Committee, 1923
International Electro-Technical Commission:
- A.S.M.E. representative on U.S. National Committee, 1923–1931
- Director of Secretariat of Advisory Committee No. 5 on Steam Turbines, 1925–1936

Chairman of Special Committee on Steam Turbines of U.S. National Committee, 1925–1936
American Engineering Council
- A.S.M.E. delegate; served from January 1, 1923, to January 1, 1926, resigning from the Coucil a year prior to the expiration of his second term because of his duties as chairman of the Boiler Code Committee
- Chairman of delegates during his presidency of the A.S.M.E. in 1924

World Power Conference, London, 1924
- President of A.S.M.E. representatives on American Committee.

One's achievements in any field are likely to be influenced not only by personal characteristics but also by early training and experience. Mr. Low was what may be termed a "self-made" man. Born in Chelsea, Mass., on April 3, 1860, he left grammar school at the age of fourteen to become a clerk in a Western Union Telegraph office in Boston where he learned telegraphy and stenography. In 1880, after two years as a court and commercial stenographer, he went with the *Boston Journal of Commerce* as secretary to the editor, Thomas Pray. This paper carried a department devoted to techni-

cal problems of the textile-mill power plants. Mr. Low studied engineering in his spare time and became intimate with operating engineers, working with them on their problems. In 1886 he was made editor of the Engineering Department of the *Journal*. It was this early experience in mastering the engineering problems of that day and explaining them to the practical men who operated the mills that accounted for his unusual ability, throughout his editorial career, to analyze and to write in a simple, direct style.

During this period with the *Journal* he married Adeline Giles, on September 24, 1881, whose algebraic problems he had labored over when she was a high-school student. He joined various engineering societies, made several power-plant inventions, and operated, in partnership with Frank M. Clark, a small company—the Clark & Low Machine Co.—for their exploitation. These inventions included an integrating steam-engine indicator, an optical indicator, an elevator control, a flue cleaner for vertical tubular boilers, and a rotary steam engine.

In 1888, Mr. Low left the *Journal of Commerce* for New York, to become the fifth editor of *Power*. This journal had been founded four years earlier, to serve the needs of power engineers in the operating field. Under Fred Low's leadership as editor, which extended 42 years, to 1930, the field of the magazine was broadened, without changing its practical approach, to serve also the needs of professionally trained power engineers, designers, and consultants.

From 1898 to 1906, Mr. Low published four technical books in the power field. In Passaic civic affairs, he was councilman (1901–1903), president of the council (1905–1906), and mayor (1908–1909).

In 1930, at the age of 70, Mr. Low became editor emeritus of *Power*. He continued to reside at Passaic, with periods spent at his farm in New Hampshire and fishing in Florida. He still maintained some contact with editorial and society work, but this gradually diminished with failing health.

He is survived by Mrs. Low, two sons, two daughters, fourteen grandchildren, and two great-grandchildren.

His deep understanding of human values and his friendly attitude were well expressed in a number of forewords which appeared in *Power*. A typical one, his New Year's Greeting for 1923, follows:

HAPPY NEW YEAR

"The old year, with all its achievements and failures, its satisfactions and disappointments, its joys and its griefs, is behind us. Its 525,600 minutes have been spent. What have we bought with them? What have we to show for it?

"The new year lies before us, a clean, unwritten page, replete with possibilities. What shall the story be which it will carry into history when, a year from now, it is turned over—into the world's history; into our life's story, yours and mine?

"The real fellow is, of course, he who plays the game the best that he knows how all the time. If he has a bad inning or a bad hole, he braces up and goes after the next with all the more concentration and purpose, and refuses to be set back by misfortune or subdued into mediocrity by temporary lack of success.

"Most of us, however, get to plodding even if we do not stumble, and we welcome a pause, a turning point in the game, a chance to size up the score and take a new hold on life. And we do this at New Year's. Today is not different from a lot of other todays; this year will not be different from a lot of other years, but we are starting in with a clean slate and a new resolve to make the most of the year that is before us and so to live that as memory turns its page it will find transcribed thereon the most satisfying accomplishment and the least of sorrow and regret.

"And for each and all I hope that the coming year will be replete with happiness and success."

Nothing could be said that would bring out his character more clearly and his knowledge of the human side than what he wrote with his own hand, and nothing would appeal more deeply to those who mourn him as a loyal coworker and a true friend.

In addition to his membership in The American Society of Mechanical Engineers, Mr. Low was a member of the National Association of Power Engineers, the Verein deutscher Ingenieure, and the Newcomen Society; also, an honorary member of the British Institution of Mechanical Engineers, the National Association of Practical Refrigerating Engineers, and the National Board of Boiler and Pressure Vessel Inspectors. His clubs included the Engineers' Club of New York, Engineers' Club of Boston, the Yountakah Country Club (Nutley, N.J.), the Plymouth (N.H.) Golf Club, and the Passaic City Club, of which he was past-president. In 1924 the honorary degree of doctor of engineering was conferred upon him by Rensselaer Polytechnic Institute.—[Memorial prepared by D. S. Jacobus, Geo. A. Orrok, and A. D. Blake, New York, N.Y., Members of the A.S.M.E.]

TRACY LYON (1865–1935)

Tracy Lyon, a member of the A.S.M.E. since 1893, died in New York, N.Y., on April 28, 1935. He was born at Oswego, N.Y., on September 13, 1865, son of James and Annie R. (Rodman) Lyon. His early education was secured at home under a tutor and he was graduated from the Massachusetts Institute of Technology in 1885 with an S.B. degree in mechanical engineering.

Mr. Lyon entered the employ of Armitage Herschell & Co. in the Tonawanda (N.Y.) engine and boiler works in September, 1886, and remained there until June, 1888, gaining a varied experience in design and construction.

A few months later he became associated with Robert Bement & Co., engineers and contractors of St. Paul, Minn. During six years spent with the firm his work dealt largely with water-works systems and hydraulic construction as far west as the Rockies. He was engineer in charge of the submarine laying of a large pipe in Duluth harbor and directed official tests of a high-duty pump. A large plant which he designed and constructed for the Walter A. Wood Harvester Co. at St. Paul included one of the early electrically operated switching services on standard gage track.

In 1894 Mr. Lyon became superintendent of motive power for the Chicago Great Western Railway, at St. Paul. He designed locomotives and cars and built the first electrically operated railway shop at Oelwein, Iowa. He became general superintendent of the road in 1899 and assistant general manager in 1903. He left in 1906 to take the position of assistant to the president, Edwin M. Herr, of the Westinghouse Electric & Manufacturing Co. He was with the company, in charge of all manufacturing, until 1911, during which period the first electric locomotives for the New Haven and Pennsylvania railroads were built by the company at Pittsburgh.

Mr. Lyon entered the automotive field in 1911 as director of production for the General Motors Company, Detroit. In 1914 he was general manager for the Cricket Cycle Car Company, and from then until 1926 engaged in operating and management work as a consulting industrial engineer in Detroit and New York. Since 1926 he had been assistant to the president of the Chrysler Corporation, New York, where his ability was highly regarded.

Mr. Lyon is survived by his widow, Frances (Gilbert) Lyon, whom he married in 1889, and by three children, Anne (Mrs. Sherman Post Haight), Laura (Mrs. Howard Calvin Sykes), and Robert Gilbert Lyon.

WILLIAM WATTS MACON (1875–1935)

William Watts Macon, former managing editor and editor-in-chief of *The Iron Age*, died at the Murray Hill Hospital, New York, N.Y., on January 1, 1935, following a cerebral hemorrhage. He had been identified with the editorial department of *The Iron Age* for 23 years.

Mr. Macon was born in New York, on May 19, 1875, the son of William F. and Harriet O. (Marsac) Macon, and was graduated from Cornell University with the degree of mechanical engineer in 1898. During the summer of 1897 he served an apprenticeship in the shops of the New York Central Railroad at West Albany, N.Y.

While at Cornell he was editor of the *Sibley Journal of Engineering* and immediately after graduation he joined the editorial department of *The Engineering Record*, with which he remained for seven years, becoming associate editor. From 1905 to 1911 he was editor of *Metal Worker*.

For some years after joining the staff of *The Iron Age* he was editor of its Engineering Department. He was managing editor from 1917 to 1930 and editor in chief from then until 1932, when he retired because of illness. Since then he had been a consulting editor to the publication. He was also a director of the Iron Age Publishing Company.

In a tribute to Mr. Macon published in *The Iron Age* for January 10, 1935, A. I. Findley, editor emeritus of the publication, said in part:

"Trained as a mechanical engineer, Mr. Macon saw early in his connection with *The Iron Age* how his horizon could be widened by close study of electrical and metallurgical developments in iron and steel and of production and management in the great metal-working field. Further broadening his equipment, he took up with enthusiasm the market side of the steel industry. And he was not long in winning a high place as an accurate reporter of market facts and a reliable interpreter of market trends.

"For years he prepared the annual survey of steel production in the different finished forms, with makers' estimates of the tonnages shipped into various channels of consumption in the 12 months. These figures, published just at the turn of the year, often came close to the official statistics compiled months later. Leading companies

were thus able to make use of them in planning for the year just ahead.

"Mr. Macon believed that contact with the men who were achieving in all metal-working lines was indispensable, if this paper were to apply its full power to building up the country's greatest manufacturing industry. He was a familiar figure at conventions and a constant student of operations at progressive plants."

In addition to his writing for *The Iron Age* Mr. Macon was a contributor to the "Encyclopedia Britannica." During the World War he was a member of the group of American trade paper journalists conducted through Great Britain and France by the British government.

Mr. Macon became a junior member of the A.S.M.E. in 1903 and an associate of the Society in 1911. He had served twice on the Executive Committee of the Metropolitan Section, the first period terminating in his chairmanship of the committee in 1919, and the second in the office of secretary-treasurer in 1930. He also served on the Executive Committee of the Iron and Steel Division from 1926 to 1931 and at the time of his death was an associate member of that committee. He had been a member of the Committee on Local Sections since 1931, and would have been eligible for its chairmanship in 1936. At the time of his death he was also serving on two special committees of the Council—the Board of Review (Delinquent Members) and the Capital Goods Industries Committee—and was a representative of the Society on the Engineers National Relief Fund and the Professional Engineers Committee on Unemployment.

Mr. Macon had also been active in the work of the American Society of Heating and Ventilating Engineers, of which he became a member in 1908, and was its secretary in 1911 and 1912. He contributed a number of papers to the society and gave constructive service to its advancement. He had belonged to the Institution of Heating and Ventilating Engineers of Great Britain, the Association des Ingénieurs de Chauffage et de Ventilation de France, and the National District Heating Association.

His other society memberships included the American Iron and Steel Institute and the American Management Association. He was secretary of his college class and was active in Cornell alumni activities, serving as treasurer of the Cornell Alumni Corporation from 1918 to 1931 and as president in 1931 and 1932. He had also been chairman of the alumni committee of the *Cornell Alumni News* and was a member and former president of the Cornell Society of Engineers and a former director of the Cornell Club of New York. Other clubs to which he belonged included the English Speaking Union and the Engineers Clubs in New York and Brooklyn. He was a member of Sigma Xi, honorary engineering fraternity, and of Sigma Delta Chi, honorary journalistic fraternity.

His widow, Mrs. Maud Andruss Macon, whom he married in 1905 at Canandaigua, N.Y., a daughter, Mrs. B. S. Cushman, and a brother, Charles F. Macon, survive him.

WILLIAM THOMAS MAGRUDER (1861–1935)

William Thomas Magruder, a member of the A.S.M.E. since 1884, and head of the Mechanical Engineering Department of The Ohio State University for almost thirty-five years, died at his home in Columbus, Ohio, on June 21, 1935.

Professor Magruder was born on April 22, 1861, at Baltimore, Md. He was the son of William Thomas Magruder (U.S.A., C.S.A.) and Mary Clayton (Hamilton) Magruder, and was a descendant of that branch of the Clan McGreagor whose members first landed in Maryland about 1662. His father was a graduate of the U.S. Military Academy, Class of 1850.

He was educated at St. John's and Peekskill Academies before entering Stevens Institute of Technology in 1878, where he won the Priestley Prize in 1880. Stevens awarded him his M.E. degree in 1881 and conferred the degree of doctor of engineering upon him in 1921.

For five years following his graduation he served as draftsman and designer for the Campbell Printing Press & Manufacturing Co., Taunton, Mass., after which he returned to college, this time to Johns Hopkins University, for graduate study in mathematics and chemistry. In 1887 he accepted the position of chief chemist of the Mt. Clare shops of the Baltimore & Ohio R.R., but resigned later in the year to become an instructor in mechanical engineering at Vanderbilt University. This was his first contact with what proved to be his major life work—engineering education. The following year (1888) he was advanced to adjunct professor of mechanical engineering. In 1896 he served as chief of machinery, Tennessee Centennial Exposition. That same year he went to Ohio State University as professor of mechanical engineering. He served this university for a total of thirty-seven years; first as chairman of the Mechanical Engineering Department until 1929, when he resigned the chairmanship but continued actively as professor in the department until he was made professor emeritus in 1933. He married Ellen Fall Malone of Nashville, Tenn., on June 18, 1891. His widow and two sons, William Thomas and Thomas Malone, survive him.

Probably Professor Magruder's major interest throughout most of his busy life centered around the problems of engineering education. The Mechanical Engineering Department at Ohio State University was comparatively young when he became its head. Its continued growth and development under his direction furnishes evidence of his ability, wisdom, and never-ending energy. His devotion to the welfare of his department, his self-sacrificing service, and his constructive activities in its behalf were unsurpassed.

He was one of the founders of the Society for the Promotion of Engineering Education and was an active worker in this organization from the time of its inception until his death. He was a councilor, 1899–1902 and 1907–1911, vice-president, 1905–1906, secretary, 1906–1907, and president, 1912–1913.

But he also maintained a large interest in his profession of mechanical engineering. His professional contacts outside the field of education were kept active, largely by means of consulting work, which occupied a not inconsiderable portion of his time until the last few years of his life. He was well known to a large number of the members of the A.S.M.E. because of the active interest he always took in the affairs of the Society. He was always willing to serve on committees, was for many years chairman of the Columbus Local Section, served on the Council and was a vice-president from 1925 to 1927. He was also a member of a number of other organizations, and served at one time or another as an officer in most of them. In the American Association for the Advancement of Science he was secretary of Section D, 1899–1900 and 1902–1907. He was president of the Columbus Engineers' Club, 1904–1905, president of the Ohio Society of Mechanical Engineers, 1905–1907, and councilor of the American Association of University Professors in 1915–1916. In 1917–1918 he was in charge of the Department of Engines, U.S.A. School of Military Aeronautics in Columbus.

He was greatly interested in the honorary engineering fraternity, Tau Beta Pi, and was largely responsible for the establishment of a chapter at Ohio State in 1921. Most of the time since then he served as a member of the Advisory Council of that chapter. From 1930 until the time of his death he was a member of the National Executive Council of that fraternity. He was also a member of Sigma Xi, Beta Theta Pi, and of the Masonic order.

Professor Magruder was a zealous member and for a large part of his life a lay-reader of the Protestant Episcopal Church. He served for many years as chairman of the Department of Religious Education for the Diocese of Southern Ohio of that church. Under his guidance this department grew from a small beginning to one which received unusual recognition.

He was the author of numerous scientific papers which were presented at meetings of societies to which he belonged or published in technical journals, and many papers dealing with educational problems, most of which may be found in the *Proceedings* of the Society for the Promotion of Engineering Education.

The present condensed biographical statement, although incomplete, will serve to give some idea as to the breadth of Professor Magruder's interests and activities and of the untiring energy which made him such a valuable member of all organizations with which he was affiliated. But it does not give any impression of his very human qualities.

He always took a great interest in people. He remembered practically all of the men who graduated from the Mechanical Engineering Department of The Ohio State University while he was at its head, and would usually call by name an old graduate who visited him, even if he had not seen him for many years. Students often came to him for advice and counsel, which he gave readily. Many of them obtained positions through his assistance which opened up opportunity for happy and useful lives.

He was a man of high standards and ideals. When he decided upon a course of action in any matter, he was not easily swayed from it, but when on the losing side of any question he would submit to the will of the majority in good spirit. His qualities of ability, energy, loyalty, and broad interests which brought him so many honors during life and made him such a valuable member of the University community and of each organization to which he belonged will indeed cause him to be sorely missed by his many friends and colleagues.—[Memorial prepared by F. W. Marquis, Columbus, Ohio. Mem. A.S.M.E.]

GEORGE HENRY MARR (1863–1934)

George Henry Marr died in Waterville, Me., on October 21, 1934, leaving his wife, Edith Jane (Stanley) Marr, whom he married in 1890, and a son, Stanley Field Marr. He had held membership in the

A.S.M.E. since 1891, being elected a junior member in that year and a member in 1905.

Mr. Marr was born at Gardiner, Me., on July 5, 1863, son of Charles Henry and Emma Augusta (Field) Marr. He was graduated from the Gardiner High School in 1882. For three years previous to graduation he passed the greater part of his vacations in the machine shop or drawing room of the firm of P. C. Holmes & Co., of Gardiner, and devoted his spare time to the study of drawing, mechanics, and the elements of machinery and machine design. Following the completion of the high-school course he was employed in the drawing room and machine shop of the company, working on the design and construction of machinery for electric-light and power plants, paper and pulp mills, and also dams, flumes, canals, etc., for manufacturing plants. He was made engineer and superintendent of the company in 1886 and continued in that position for ten years.

From 1896 to 1900 Mr. Marr was connected with the Stillwell-Bierce & Smith-Vaile Company of Dayton, Ohio, his duties including the survey, design, and construction of hydroelectric power plants and general mill work. During the last year of this period he was New England agent for the company, with headquarters at Boston.

From 1900 until his retirement thirty-three years later, Mr. Marr was mechanical engineer for the Hollingsworth & Whitney Co., of Waterville. He designed and superintended the construction of paper and pulp mills, power plants, and equipment. He patented a pneumatic diaphragm pulp machine.

Mr. Marr was also interested in natural science. He was a member of the American Microscopical Society and the Cambridge Entomological Society.

He served as secretary of the Waterville Rotary Club for ten years, was secretary of the Waterville Historical Society, and member of the City Planning Board, and trustee and secretary of the Public Library.

BENJAMIN MOODIE MAYNARD (1880–1934)

Benjamin Moodie Maynard was born at Roxbury, Boston, Mass., on December 20, 1880, the son of Benjamin and Mary Teresa (Dolan) Maynard. He attended the Roxbury Latin School and was graduated from Harvard University in 1904 with an A.B. degree.

After six years of what he referred to as "investigation work" for the Cabot Manufacturing Company, Remington Typewriter Company, and others, he became connected in 1910 with Miller, Franklin, Basset & Co., New York. He held the position of junior engineer for a year and a half and that of senior engineer for an equal period of time. He was then advanced to the position of supervising engineer, directing and planning installation work of a group of senior engineers.

From 1917 until he left the firm in 1925 Mr. Maynard had full charge of all of its industrial engineering work. This entailed the training and supervision of a large group of engineers engaged in this work and full responsibility for the methods used. He originated and developed the theory and practice of time-study technique used by the firm and originated several phases of production planning, notably the solution of the problem of "when to start" an order or lot. He installed or supervised the installation of costs, production-planning, wage-incentive, time-study, and other methods in hundreds of plants, and laid out and in some cases specified mechanical and tool equipment in many shops.

Shortly before he left the firm the following tribute by William R. Basset, then head of the organization, was printed in "The Sphinx Talks," published by the company.

"From the sublime to the ridiculous—or so it seems. From work which took him amid thousands of whirring spindles, humming machine tools, and ponderous paper machines, he has turned to baking cake!

"Benjamin Moodie Maynard came with us in 1910, possessed of a Harvard education, six years' experience as a lace salesman, an easy personality, and a fund of common sense.

"Few men possess a more diversified business experience. We are printing herewith the names of the companies for whom he worked in our behalf. The list is probably not complete. It has already been twice revised to include names we had forgotten. In some cases he spent months in the factories and offices named—though, in the later years, he was used largely upon the briefer investigations and in performing the periodic supervision given all our field men.

"Several years ago he planned and financed a small company which undertook to bake cake for distribution to grocers, lunchrooms, and others. His wide experience made success a foregone conclusion. His rapidly growing cake bakeries—there are five already, and there will be more—demand almost all of his time. Therefore he is gradually withdrawing and is planning to complete his withdrawal before 1926 is ushered in.

"We will miss him. We already do miss him."

The list of companies referred to by Mr. Basset cannot be reproduced here, but it shows a wide diversity in the size and production of the plants which Mr. Maynard served.

Mr. Maynard died suddenly on April 14, 1934, while on his way to New York from his home in Old Greenwich, Conn. He is survived by his widow, Angelica (Dampf) Maynard, and by two children, Mary and Jane.

He had been a member of the A.S.M.E. since 1921.

JOHN BOYLSTON MAYO (1859–1934)

John Boylston Mayo, retired, a resident of East Orange, N.J., died of heart disease at the Homeopathic Hospital there on November 29, 1934, shortly after his seventy-fifth birthday. A son of John Stratton and Olive Vining (Vinton) Mayo, he was born on November 25, 1859, at Worcester, Mass., and secured his education in the public schools of that city.

He served an apprenticeship as carpenter, millwright, and patternmaker with the Washburn & Moen Mfg. Co., Worcester, from 1877 to 1882, and during the next four years was engaged in patternmaking for that company, the Holyoke Machine Company, Morgen Spring Company, and the Pond Machine Company, Worcester. He then went to Anniston, Ala., where he was employed for three years as draftsman and superintendent of construction of blast furnaces for the Woodstock Iron Company.

He returned to New England in 1889, and was employed as a draftsman first by the Whittier Machine Company of Boston, and then by the Yale & Towne Manufacturing Co., Stamford, Conn. Subsequently he went to New York to work for the Link-Belt Engineering Company and to Chicago where he was employed by the Crane Elevator Company. He returned to New York to take a position as draftsman with Otis Brothers & Co., where he was working when he joined the A.S.M.E. in 1892. Two years later he was chief draftsman for the Crocker-Wheeler Electric Manufacturing Company, Ampere, N.J., and later he served that company as checker. Other companies for which he worked as chief draftsman or checker were the Coe Brass Manufacturing Company, Torrington, Conn., Alpha (N.J.) Portland Cement Company, Texas Portland Cement Company, Cement, Tex., New London Ship & Engine Co., Groton, Conn., Southwark Foundry & Machine Co., Philadelphia, Pa., and Public Service Production Company, Newark, N.J. He was a frequent contributor to engineering periodicals and belonged to the Masonic fraternity.

Mr. Mayo was twice married, his first wife being Margaret Atkin, whom he married in 1895, and who died in 1913, and the second, Ida Kenny, whose death occurred in 1918, three years after their marriage. He is survived by two sons, John B., Jr., and Albert R. Mayo.

GEORGE FREDERICK McCLELLAN (1873–1934)

George Frederick McClellan was born at Newark, N.J., on October 21, 1873, his parents being George F. and Julia Beulah McClellan. He supplemented his early schooling with special work at Stevens Institute of Technology and correspondence-school courses.

After some time with the Crocker-Wheeler Electric Manufacturing Company, Ampere, N.J., Mr. McClellan was chief engineer from 1900 to 1905 for the Weston Electrical Instrument Company, Newark. He then became master mechanic for the Public Service Railway Company, Newark, where he remained until 1911. During the next two years he was works engineer for the Whitehead & Hoag Co., Newark, for whom he designed special machinery for the manufacture of novelties and had general charge of all machinery, power transmission, and buildings.

In 1913–1914 Mr. McClellan was engaged in the reorganization of the operating force and the supervision of installing new units and repairing existing machinery for the Lakewood & Coast Electric Co., Lakewood, N.J. The following year was devoted to general engineering work under his own name, specializing in power-plant efficiency. This led to the position of chief engineer for the Hyatt Roller Bearing Company, Harrison, N.J., where he remained two years.

When Mr. McClellan became a member of the A.S.M.E. in 1918 he had but recently become inspecting engineer of public utility properties for Henry L. Doherty & Co., New York, the position in which he continued until his death on September 17, 1934. He also carried on exhaustive studies in transportation for that company.

Mr. McClellan was a member of the National Guard of the State of New York, was a licensed professional engineer in that state, and had been a member of the National Electric Light Association. He also belonged to the Masonic fraternity. He is survived by his second wife, Sarah May (Richter) McClellan, whom he married in 1926, and by seven daughters by a previous marriage.

JOSEPH ALOYSIUS McELROY (1859–1934)

Joseph Aloysius McElroy, treasurer of the construction engineering firm of McElroy & Kerwin, New York, N.Y., though not active in the business for some years, died at the hospital in Norwalk, Conn., where he made his home, on April 28, 1934, after a brief illness. Prior to his virtual retirement at the close of the World War he had become a leading engineer in street-railway construction in various parts of the world.

Mr. McElroy was born in Bridgeport, Conn., on March 20, 1859, the son of Charles and Margaret McElroy. Circumstances necessitated his going to work after he had left public school. He served an apprenticeship as a machinist at the Coulter & McKenzie Machine Co., Bridgeport, Conn., from June, 1877, to June, 1880, and during the following three years worked as a journeyman in Providence, R.I., and Boston, Mass., about a year of the time with Brown & Sharpe, in the former city, and the remainder with the Boston Machine Company, Waltham Watch Company, Ashcroft Steam Gauge Company, and Boston Sewing Machine Company. During these years he studied nights to prepare himself for college, and in 1887 he was graduated from Stevens Institute of Technology with a mechanical engineering degree.

After Mr. McElroy completed college his first position was with the United Gas Improvement Company, of Philadelphia, Pa., and in their interests he went to Omaha, Neb., as assistant superintendent of gas works. In 1888 he went to Cuba as engineer for J. Rigney & Co., on a sugar estate. The following year he returned, and for a time was superintendent of the machine shop and foundry for Charles Reed, of Danbury, Conn., manufacturer of hatter's machinery. In 1890 he entered the employ of the Field Engineering Company, New York, N.Y., and for several years was in charge of its construction work in western New York. He then became associated with Thomas Murray, New York, and engaged in street-railway construction in Bridgeport and in New Jersey.

Mr. McElroy became greatly interested in this line of work and formed a partnership with John F. McCartney under the firm name of McCartney and McElroy for contracts in that field. The firm built lines in Hamilton, Ontario, Highland Falls, N.Y., and from Hoosick Falls, N.Y., to Bennington, Vt. After a few years Mr. McCartney went abroad to investigate the possibility of contracts there. He secured one in Glasgow, Scotland, and Mr. McElroy soon joined him in London, where they incorporated a company under the name of McCartney, McElroy & Co., Ltd., with London as headquarters. Mr. McElroy remained abroad for about ten years, personally supervising construction of the first electric tramways in Glasgow and Manchester, England, and in Lisbon, Portugal. He also had charge of construction of the first street railways in Aberdeen, Scotland; Brighton and Stockport, England; Durban and Port Elizabeth, South Africa; Wellington, New Zealand; and on the Island of Malta. The company also owned and operated the train and freight-elevator system at Malta for a few years.

After his return to the United States Mr. McElroy located in Norwalk, where he secured paving contracts. He built a home there and organized the Empire Construction Company. Much of the work carried out by this company was in New York, and included the tracks for the Lexington Avenue subway from Woodlawn to 38th Street and extensions to the Liberty Avenue and Myrtle Avenue lines of the Brooklyn Elevated Company, now part of the Brooklyn-Manhattan Transit system. The present firm of McElroy & Kerwin was organized shortly after the World War, but as already stated, Mr. McElroy's connection with it was chiefly as treasurer.

Much of his time during recent years had been given by Mr. McElroy to community interests in Norwalk. He was named a member of a committee to let contracts for a sewage disposal plant there and during the construction of the plant served on a special supervising committee. He was subsequently named a member of the Sewage Disposal Commission, on which he served until within a few months of his death. He also served as chairman of the Board of Directors of the Norwalk Public Library. He was a trustee of the Knights of Columbus and a member of the Holy Name Society of St. Mary's Church, in Norwalk. He had been a member of the A.S.M.E. since 1895.

Mr. McElroy is survived by his widow, Alice (Dial) McElroy, and by three children, Wilber J., Paul C., and Alice E. McElroy.

WALTER MARTIN McFARLAND (1859–1935)

Captain Walter Martin McFarland, past vice-president of the A.S.M.E. and past-president of the Society of Naval Architects and Marine Engineers, died at his home in Washington, D.C., on March 4, 1935.

Captain McFarland was born in Washington on August 5, 1859, the son of John and Sarah J. (Slater) McFarland. He attended the public grammar schools of Washington and was graduated at the head of his class. After winning a Kendall Scholarship in competitive examination, he entered the Columbian University Preparatory School in Washington. He attended this school only one year as he won, also by competitive examination, an appointment to the Naval Academy in 1875 as a cadet engineer. He was graduated in 1879, standing second in his class.

Following graduation he was detailed for several months on the practice tug *Standish*, after which, from October, 1879, to November, 1881, he served at sea in American and European waters as a junior engineer on board the U.S.S. *Nipsic* and the cruiser frigate *Trenton*. During the early part of 1882 he was on special duty in the drafting room of the Bureau of Steam Engineering. He was then promoted to assistant engineer and detailed to the U.S.S. *Michigan*, later known as the *Wolverine*, where he served on the Great Lakes until 1883.

In June, 1883, he was detailed to Cornell University, where he served as assistant professor of mechanical engineering until 1885. His success there was so great that the president of Cornell, Andrew J. White, requested the Navy Department to extend his detail. This, however, was not granted, and he was ordered to inspection duty at the Morgan Iron Works for a year. From 1886 to November, 1888, he was detailed to sea duty and served in Pacific waters on the U.S.S. *Vandalia*. This sea duty left a lasting impression on young McFarland and formed the background for many of the stories and anecdotes for which he was noted.

In December, 1889, he was detailed to the Bureau of Steam Engineering as one of the assistants to the engineer-in-chief, Admiral George W. Melville. Ever alert to surround himself with bright young men, Admiral Melville noticed that McFarland possessed tact and the ability to make quick decisions tempered with sound judgment. This discovery resulted in his being made confidential assistant to Admiral Melville and he acted as personal representative of Admiral Melville on many important missions.

In 1889, while attached to the Bureau, he took an active part in the organization of the American Society of Naval Engineers and in April, 1890, was elected secretary-treasurer of that society and editor of its journal. From the inception of that journal he contributed many articles of technical value. His published methods of speed determination by the number of revolutions of the propeller were among his greatest contributions to marine engineering at this stage of his career.

He served as secretary of the Division of Marine and Naval Engineering and Naval Architecture at the International Engineering Congress which met at the World's Columbian Exposition at Chicago in 1893. This Congress was attended by representatives and delegates from the technical societies in England, France, Germany, Italy, Japan, Spain, and Norway. As a result of his work this Congress publicly commended him for his alert mind, ability, and personality. At this time he won the admiration and respect of some of the outstanding delegates from these countries which resulted in lifelong friendships.

In September, 1891, he was promoted to past assistant engineer and continued to serve as confidential assistant to Admiral Melville until 1894. After being detached from the Bureau of Steam Engineering he delivered a series of five lectures on marine engineering at the Naval War College at Newport before reporting for sea duty on the U.S.S. *San Francisco*. He served on this ship in European waters until July, 1897. In the summer of 1897 he represented the United States Navy Department at the International Congress of Naval Architects and Marine Engineers held at London, England.

He was then detailed again to the Bureau of Steam Engineering and served as assistant to Admiral Melville until July, 1899. While serving at the Bureau he was, in 1898, promoted to chief engineer.

When he reported at the Bureau of Steam Engineering in 1897 the old controversy between the Line and the Engineer Corps of the Navy was a subject of special consideration. Up to this time conflict had been waged between these two branches of the Service for many years. Engineer cadets, upon graduation from the United States Naval Academy, were not given military rating. They were merely commissioned engineers who were without power to command enlisted men.

In writing on this subject at the time, Admiral Melville said:

"Under a naval organization which was based on the needs of days when steam was only an auxiliary, but which has continued to the present, it could not be otherwise than that the varying interests of Line and Engineer officers, with their widely distant points of view, should produce the conflict which has been waged between these branches of our Navy for more than thirty years. Discord was inevitable—the friction between the old order, fated to pass away, and the engineer, who, in great degree, represented that which was

to succeed it. The Line officer, justly proud of the past achievement of his corps, was disposed to yield none of his military functions to the newcomer; while the Engineer, trained in the same naval school and side by side with his brother of the Line, could not but see injustice in a system which gave him a vast responsibility in the care of the machinery of all types on the modern man-of-war, with the control, in some cases, of half her crew, and yet refused him the right to command enlisted men, classed him, despite his equal danger in action, as a noncombatant, and denied him the military title which, through centuries, has been associated with the fighting man.

"Many able and patriotic men, within and without the Navy, recognizing the danger to a military service whose house was thus divided against itself, have, time and again, made efforts, until recently in vain, to find a solution of the problem which would reconcile all differences."

In November, 1897, Secretary of the Navy Long appointed a Personnel Board to straighten out the friction between the two Corps. Theodore Roosevelt, who was then Assistant Secretary of the Navy, was the presiding officer of this Board, which consisted of the following: Line Officers Crownshield, Evans, Hemphill, Key, McCormick, Sampson, and Wainwright, and Engineers Kearney, McFarland, Melville, and Rae.

The engineers demanded three things: First, the right to exercise military command over men in the Engineering Department; second, the legal right to command any enlisted man; third, an engineering rank coupled with a real, not relative, military title to prevent confusion with the Line officers.

At the first meeting of the Board, Rear-Admiral, then Captain, Robley D. Evans, suggested that the Line and Engineer Corps be amalgamated.

Past Assistant Engineer McFarland, who had the honor of acting as the sole sponsor for the younger men of his Corps, was from the first one of the most active and efficient supporters of the amalgamation scheme. He was a fluent speaker and, in spite of his junior years, did most of the talking for the engineers. At the close of the first day's conference where the amalgamation scheme had been proposed, the Chairman of the Board, Assistant Secretary Roosevelt, asked Engineer McFarland to remain. In the course of their conversation, Mr. Roosevelt said: "I am in sympathy with the claims of your side but unfortunately there is no precedent." To this McFarland replied: "Mr. Secretary, you probably remember how, in the play of Madame Sans Gene, one scene shows an evening in Napoleon's court where conversation turned on ancestry and descent, and he was asked: 'Who was your ancestor?' Napoleon touched his breast and replied: 'I am the ancestor.' Mr. Secretary, we will make the precedent." Anyone knowing Mr. Roosevelt's love for establishing precedents can well imagine the impression McFarland's remark made on the then future President.

It is interesting to note the statement made by this Board in reporting favorably on the amalgamation proposal:

"Every officer on a modern war vessel in reality has to be an engineer whether he wants to or not. Everything on such a vessel goes by machinery and every officer whether dealing with turrets or the engine room has to do engineer's work.

"In making this change we are not making a revolution; we are merely recognizing and giving shape to an evolution, which has come slowly, but surely and naturally; and we propose to reorganize the Navy along the lines indicated by the course of the evolution itself."

During the hearings on the bill before the Naval Affairs Committee, Engineer McFarland proved a powerful advocate of the measure. His alertness of mind, his courage, and his knowledge of all phases of the subject were important factors in obtaining the bill's approval by the Committee and drew from the Chairman of the Committee the comment that, "He was the best posted man that they had ever examined."

The Personnel Bill was enacted by Congress in 1899 and the engineers obtained military rank. The precedent for military rank, which all the corps in the Navy now have, was thus established.

During his work at the Bureau of Steam Engineering under Admiral Melville, Chief Engineer McFarland's duties brought him in contact with many prominent men, such as George Westinghouse. Like Admiral Melville, Mr. Westinghouse was attracted to Captain McFarland and the outcome was an offer of a vice-presidency with the Westinghouse Electric & Manufacturing Co. He accepted this position and resigned from the Navy, effective July 5, 1899. In commenting on the loss of his services, Admiral Melville said: "While I cannot but congratulate my good friend George Westinghouse, whose name is a synonym for engineering genius and business enterprise, on McFarland's entrance into his official corps, I cannot but regret the loss to the fleet of one of its brightest men and to myself of a most talented and loyal assistant."

Captain McFarland remained in the employ of the Westinghouse Electric & Manufacturing Co. until 1910, when he resigned to become manager of the Marine Department of The Babcock & Wilcox Co., thus reentering the field of steam engineering for which he had earlier shown great aptitude and attachment.

A news item in *Electrical World* for March 31, 1910, gives the following résumé of his ten years as vice-president of the Westinghouse Electric & Manufacturing Co.

"He has had official supervision of the large contracts of the company, as well as being the advisory head in all the cooperative movements of the company with the associated Westinghouse Companies involving literature, advertising, and exhibition work. As a frequent representative of the Westinghouse Companies at important meetings of engineering societies and at conventions, he is well known, and has long been looked upon, by his company at Pittsburgh, as the official host. In the latter capacity he has come into contact with many distinguished engineers and other guests from all parts of the world. His broad experience in engineering matters in general, attained through his previous work in the United States Navy, and his personal acquaintance with men of public affairs, entirely fitted him for duties of this character. In his connection with the Westinghouse Electric & Manufacturing Co. he has done much to systematize and improve the work of the departments with which he has come into contact and has, through his personal qualities, won the confidence and respect of the large number of employees looking to him for guidance and direction."

During his services with The Babcock & Wilcox Co., he cooperated in the development of marine boilers and particularly in the utilization of higher pressures and temperatures. Oil-burning apparatus, which so revolutionized boiler operation in the Navy, was developed and perfected during this time. Under his management, the company built the greater part of the boilers installed in United States naval and merchant vessels during the World War.

Captain McFarland retired from active service on September 1, 1931, on account of poor health. He then took up his residence in Washington, D.C., his old home, where he lived until his death.

Shortly before his retirement, Captain McFarland's own wide store of common sense and his unusual ability to draw conflicting ideas and interests together resulted in his unanimous choice for the chairmanship of the Committee to Coordinate Marine Boiler Rules. The work of this committee is of far-reaching importance and is one of the most outstanding contributions to marine engineering of recent times.

At the suggestion of Secretary Lamont, at that time Secretary of Commerce, this committee was organized on December 27, 1929, for the purpose of coordinating the Boiler Code Rules adopted as standard by the American Marine Standards Committee and those tentatively promulgated by the Steamboat Inspection Service. The committee was formed by requesting The Society of Naval Architects and Marine Engineers, the American Bureau of Shipping, the National Council of American Shipbuilders, the Steamboat Inspection Service, The American Society of Mechanical Engineers, the American Steamship Owners' Association, and the American Marine Standards Committee to appoint representatives on the committee. All responded by appointing a representative on the committee, which was later known as the Committee to Coordinate Marine Boiler Rules.

After the formation of the committee it was agreed that an outside man should be appointed as chairman. The unanimous choice fell on Captain McFarland. He was an ideal chairman, his attitude in conducting the meetings being well expressed in a preamble which he prepared for the report of the committee, a paragraph of which read as follows:

"A guiding principle from the beginning has been that the code should represent the unanimous opinion of the committee. In other words, if differences of opinion arose, they were to be smoothed out and settled by unanimous vote. With great satisfaction the committee can assert that this has been accomplished."

Captain McFarland became a member of the A.S.M.E. in 1883. In addition to his term as a vice-president in 1905–1907 he served on several committees of the Society, including the Nominating Committee (1909), Library Committee (1911–1919), and Special Committee on Flanges (1911–1914).

He became a member of the Society of Naval Architects and Marine Engineers in 1893, the year it was founded. He was elected a member of its council in 1897, a vice-president in 1918, and president for the years 1922–1924. In 1927 he was made an honorary member of the society.

In April, 1913, Captain McFarland was elected a trustee of Webb Institute of Naval Architecture. His previous experience at Cornell University caused his appointment on the Education Committee. A few years later he was elected chairman of the Committee. Upon the resignation of the vice-president of Webb Institute in 1917, he was elected to serve in that position as well as chairman of the Education Committee. Upon the death of Mr. Stevenson Taylor in 1926,

Captain McFarland was elected president of Webb Institute, which office he held until his retirement in 1931, when he was made president emeritus.

During his lifetime, Captain McFarland achieved many honors and had an unusually wide circle of warm friends. He was particularly considerate of younger men, to whom he was an inspiration.

Until the time of his death, he took a keen interest in all matters concerning shipbuilding, engineering, and the Navy. In 1928 he served as a member of the Board of Visitors of the Naval Academy and later was president of the Naval Academy Graduates Association of New York. For many years he was most active in the Engineers' Club of New York.

He found time to write many articles for technical journals and to lecture on engineering and economics subjects at Columbia University, Cornell University, Johns Hopkins University, and the Postgraduate School of the United States Naval Academy. He was particularly noted for his unusual gift of common sense and this with his technical ability and remarkable memory as well as his facility in the expression of keen, clear, and vigorous English were well exemplified in his lectures and writings.

The death of Captain McFarland is a great personal loss to his life-long friends and associates and a great loss to the societies which he served so devotedly with ability and distinction. His long record of a useful and distinguished life, specially marked by regard for his fellow men and devotion to their interests, will serve as an example to those who knew him and his memory will be cherished by all of those who had the good fortune to be numbered among his friends.—[Based on a memorial prepared by J. H. King, Manager of the Marine Department of The Babcock & Wilcox Co., New York, N.Y., for the Society of Naval Architects and Marine Engineers.]

MAINOR STUART MELVILLE (1883–1935)

Mainor Stuart Melville, works manager for the Benedict Manufacturing Company at East Syracuse, N.Y., died suddenly at his home in Dewitt, N.Y., on April 25, 1935. His widow, Beatrice (Giltner) Melville, and two sons, Thomas and John Stuart Melville, survive him.

Mr. Melville was born at Ballston Spa., N.Y., on August 19, 1883, son of Thomas and Martha (Benedict) Melville. He attended Ohio Northern University for a year and was graduated from Cornell University with an M.E. degree in 1912. He was connected with the Benedict Manufacturing Company from that time until 1919, for the first two years in the Cost Department, and during the remainder of the time in charge of the Engineering and Cost Departments. The company manufactures silverware and the engineering work involved was in connection with the design and application of new equipment for that purpose.

From 1919 until the spring of 1931, Mr. Melville was with the Oneida Community, Ltd., Oneida, N.Y., for two years as mechanical engineer engaged in research and engineering in connection with the development of new machines for the silverware business, then in charge of the Engineering and Power Departments, designing and developing new machinery, and beginning in 1926, in the capacity of advisory engineer for the firm. He had been works manager of the Benedict Manufacturing Company since May 1, 1931.

Mr. Melville became a member of the A.S.M.E. in 1928, and belonged to the Technology and Citizens clubs in Syracuse.

ALBERT SIDNEY MERRILL (1878–1935)

Albert Sidney Merrill was born in Auburn, Me., on July 3, 1878, the son of Frank P. and Elizabeth (Ring) Merrill. After graduation from the high school at Malden, Mass., where the family moved when he was about ten years old, he entered the Massachusetts Institute of Technology. He was graduated in 1900 with an S.B. degree and during the following year was an assistant in mechanical engineering at the Institute. He was engaged in field and office work for the Massachusetts Highway Commission in the summer of 1901, and then went to Madison, Wis., to serve as instructor in mechanical engineering at the University of Wisconsin. He taught there two years, spending the intervening summer vacation as draftsman at the McCormick Works, in Chicago, of the International Harvester Company.

At the close of the school year in 1903 Mr. Merrill went to Washington, D.C., to take the position of assistant physicist at the Bureau of Standards, in charge of the Materials Section. He was there until May, 1906. From then until 1910 Mr. Merrill was located in Chicago. He was an instructor in mechanical engineering in the evening school at Lewis Institute during the entire period, and also, from 1906 to 1908, salesman for the Sullivan Machinery Company, and from 1908 to 1910, assistant inspecting engineer for the Universal Portland Cement Company.

In the fall of 1910 Mr. Merrill joined the faculty of Lafayette College, at Easton, Pa., as instructor in civil engineering, in charge of the Materials Testing Laboratory and classes in applied mechanics, mechanics of materials, and reinforced concrete. Two years later he again took up the dual role of teaching and commercial work, being instructor in civil engineering at the Cooper Union Evening School, New York, and assistant to the chief engineer of the Turner Construction Company. He resigned the former connection early in 1914 but continued with the Turner company until the United States entered the World War.

He was commissioned a first lieutenant in December, 1917, and was assigned to the Inspection Division of the Ordnance Department and detailed for duty at a munitions plant in Toronto, Canada. He was discharged from service in November, 1919.

Following his release from the Ordnance Department Mr. Merrill took a position as consulting concrete engineer at the Bureau of Standards. Toward the end of 1920 he became a member of the engineering organization of the Westinghouse Lamp Company at Bloomfield, N.J. He was transferred to the Belleville, N.J., plant of the company in November, 1923, and served as office manager there until his retirement from business in July, 1926.

Since his retirement Mr. Merrill had been greatly interested in antiques and in compiling genealogical records. He had data nearly complete for a genealogy of the Ring family.

He died at his home in Auburn on March 31, 1935. He is survived by his widow, Amy Attwood (Wilder) Merrill, whom he married in 1930, and by a step-daughter and a step-son, Venetia and William A. Wilder, of Auburn.

Mr. Merrill became a junior member of the A.S.M.E. in 1903 and a member in 1914. He belonged to the Phi Gamma Delta fraternity and to the Universalist Church in Auburn.

JOHN ADAM MILLER (1871–1934)

John Adam Miller was born at Ossining, N.Y., on August 11, 1871, son of William and Margaret (Griffiths) Miller. He attended the Ossining public schools, served an apprenticeship in the plumbing trade from 1886 to 1890, and studied mechanical drawing at Cooper Union in 1888–1889. After some years as journeyman and foreman in heating and plumbing work he became a senior partner, in 1899, of Miller Bros., Tarrytown, N.Y., where he remained until 1916, specializing in sheet-metal work.

From 1916 to 1920 Mr. Miller was connected with the Efficiency Department of the Chevrolet Motor Company, specializing in the design, construction, and erection of ovens, conveyors, and equipment used in auto-assembly plants and supervising installations in St. Louis, Mo., Muncie, Ind., Flint, Mich., and other cities. He was in charge of the department in 1919–1920.

In 1920 Mr. Miller became president of C.M.S., Inc., of Tarrytown, the name of which was changed to Miller-Somes, Inc., in 1931. He served in this capacity and as treasurer of the company until his death on December 3, 1934. The company handles the design, construction, and erection of equipment for auto-assembly plants, including electrically heated painting and enameling ovens, and manufactures portable electric heaters.

Mr. Miller became a member of the A.S.M.E. in 1922, and belonged to the Odd Fellows. He married Miss Adelaide Dowling in 1916 and is survived by her and by four children, William, James, Robert, and Margaret Miller.

FRANKLIN MOELLER (1865–1935)

Franklin Moeller, manager of the New York office of The Wellman Engineering Company, died at Stroudsburg, Pa., on February 23, 1935, as the result of an automobile accident. He was born in New York on April 12, 1865, son of Peter W. Moeller, principal of the Moeller Institute, a school for boys. He attended the United States Naval Academy at Annapolis for a year and a half, then entered the Stevens Institute of Technology, from which he was graduated in 1887. He ranked high in his studies and although he missed some of the extra curricular activities because he lived in New York, he was a favorite of the class, known affectionately by the nickname "Maud," which he had brought with him from Annapolis.

Following his graduation, Mr. Moeller worked for short periods for Johnson & Morris, steam heating engineers of New York, and for the Welsbach Incandescent Gas Light Company, New York, as an assistant engineer. He then became a draftsman for the Ingersoll-Sargent Rock Drill Company, New York, with whom he worked until 1890.

At that time he entered the employ of the Webster Camp & Lane

Machine Co., Akron, Ohio, as a draftsman. Later he became chief draftsman and assistant to the general manager. In 1894 he went to Providence, R.I., to serve as mechanical engineer for the William A. Harris Steam Engine Co. and in 1896 to Brooklyn as designer for the Guild & Garrison Steam Pump Works. He then returned to Webster Camp & Lane Machine Co. in Akron. In 1903 the company was consolidated with The Wellman-Seaver-Morgan Engineering Company of Cleveland, Ohio, through the incorporation of the two companies under the new name of The Wellman-Seaver-Morgan Company, and Mr. Moeller then went to Cleveland with the new company.

A few years later he became engineer in charge of the Mining Machinery Department, and contributed much to the development of both the large steam-driven and electric-driven mine hoists, and other mining machinery built by this company.

In 1917 he was made manager of the Foreign Sales Department and continued in this position until April 15, 1925, when he was transferred to New York as the manager of the office of The Wellman-Seaver-Morgan Company (the name of the company was changed in 1930 to The Wellman Engineering Company), in which position he continued until his death.

Mr. Moeller became a junior member of the A.S.M.E. in 1891 and a member in 1912. He also belonged to the American Institute of Mining and Metallurgical Engineers, the Cleveland Engineering Society, the Engineers' Club of New York, and the Society of Terminal Engineers, of which he was a recent vice-president.

In 1894 Mr. Moeller married Isabelle W. Beck, of New York, who died in 1928. They are survived by a son, William A. Moeller, and by two daughters, Mrs. Sheldon K. Towson and Mrs. Edwin D. Veldran, of Oradell, N.J.

WILLIAM OTIS MOODY (1868–1934)

William Otis Moody, mechanical engineer for the Illinois Central Railroad Company, died at his home in La Grange, Ill., on December 25, 1934.

Mr. Moody was born in Chicago, Ill., on January 12, 1868, son of Andrew and Sarah (Clark) Moody. After being graduated from the Chicago Manual Training School, he began work as stationary engineer for the St. Francis (Arkansas) Lumber Company. Between 1890 and 1893 he served an apprenticeship as a machinist at the Weldon Shops of the Illinois Central Railroad and worked for a time on the construction of steel passenger cars. In 1893, during the World's Fair (Columbian Exposition) at Chicago, he was with the Root Porative Pressure Blower Company. He also worked for a time as draftsman for Fairbanks, Morse & Co.

From 1894 to 1896 Mr. Moody was engine designer for the Gates Iron Works, Chicago, and since then had been connected with the Illinois Central system. He was appointed chief draftsman in May, 1896, and ten years later was made mechanical engineer, the position he had since held.

Mr. Moody became a member of the A.S.M.E. in 1907. He was also a director of the Railway and Locomotive Historical Society of Boston, Mass., and a member of the Masonic fraternity. He was particularly devoted to tennis and played until within about a year of his death. He is survived by his widow, Agnes Daisy (Deane) Moody, and by two sons, John Otis and William Deane Moody.

WALTER EDMUND JOSEPH MOORE (1892–1934)

Walter Edmund Joseph Moore, who died on October 1, 1934, was vice-president in charge of operations of The Quaker Maid Co., Inc., New York, N.Y. He had been with this company and its predecessor, the A & P Products Corporation, since March, 1917, with the exception of two short periods. From May, 1917, to the end of 1918, he was a corporal in the Chemical Warfare Service of the United States Army, stationed at Edgewood Arsenal in Maryland, and for about a year in 1924–1925 he was a partner in the firm of Peters & Peters, Inc., New York, engineers and contractors specializing on changing over steam-driven plants to motor drive.

Son of John A. and Mary A. (Killen) Moore, Mr. Moore was born on April 2, 1892, in Jersey City, N.J., where he resided at the time of his death. He was graduated from the Jersey City High School and in 1915 from Stevens Institute of Technology, Hoboken, N.J., where he secured a mechanical engineering degree. During his vacations and for about six months after graduation he worked in the engineering department of the Jersey City plant of the American Sugar Refining Company. He then entered the employ of the Brooklyn Union Gas Company, in which he rose to the position of district superintendent in charge of its largest pipe-fitting shop and warehouse.

Prior to 1924 Mr. Moore was assistant chief engineer of the Great A & P Tea Company, in complete charge of the maintenance and equipment of the headquarters plant in Jersey City and cooperating in the company's outside work, comprising new buildings and manufacturing equipment. He became engineer for the A & P Products Corporation at Brockport, N.Y., in 1925, and was transferred to the headquarters office in New York in 1926, and in 1931 was made vice-president in charge of operations.

Mr. Moore became a junior member of the A.S.M.E. in 1918 and a member eight years later. He also belonged to the Sigma Nu fraternity. He is survived by his widow, Helen Grace (Hanley) Moore, whom he married in 1923.

ANTHONY SAUNDERS MORRIS (1862–1935)

Anthony Saunders Morris was born in Philadelphia, Pa., on April 13, 1862, the son of Henry Gurney and Sallie (Marshall) Morris. He attended the Episcopal Academy and Penn Charter School in Philadelphia and was graduated from Stevens Institute of Technology with an M.E. degree in 1884. During the next two years he was employed by Henry G. Morris, Philadelphia, as draftsman and designer for sugar and gas machinery. He then became superintendent for the Julien Electric Company, Camden, N.J., which was engaged in the development of American methods of manufacturing storage batteries and patents of Edward Julien, of Belgium.

In 1887 Mr. Morris took up the student shop course of the Westinghouse Electric Company at Pittsburgh. Later he served six months as superintendent of the company's electric-light plant at New Orleans, La., and then as assistant in the laboratory, working on transformer and meter design. He left the Westinghouse company in 1891 to enter the employ of the Edison General Electric Company, of Boston. He was sent to work on alternator and transformer design at its Cleveland Works, the Brush Electric Company, and spent nearly a year there.

Returning to the Westinghouse company in 1892, Mr. Morris was assigned to the Power Transmission Department. He later worked on the staff of the general manager as correspondent in charge of the Polyphase Department, handling all orders, price lists, and instructions to district officers for the department. In 1898 he was commissioned sales engineer, with headquarters in Philadelphia.

Mr. Morris remained with the Westinghouse company until 1905, when with his brother he formed the P. H. & A. S. Morris partnership of Philadelphia for manufacturing sugar machinery, especially centrifugals and evaporators. The name of Morris Engineering was adopted two years later and in 1912 the power transmission business of the George V. Cresson Company was taken over and the Cresson-Morris Company formed. Mr. Morris served as vice-president of this company, as he had of Morris Engineering. Subsequent to 1907 he was also secretary and chief engineer of the Kestner Evaporator Company, designing and manufacturing evaporators. He retired in 1931.

Mr. Morris became a member of the A.S.M.E. in 1922 and also belonged to the American Institute of Electrical Engineers, The Franklin Institute, the Theta Xi fraternity, and several clubs, including the Rittenhouse in Philadelphia and the Merion Cricket in Haverford, where he resided. He died on April 12, 1935, his skull being fractured in a fall downstairs at his home. He is survived by his widow, Mrs. Elisabeth H. (Wood) Morris, whom he married in 1890, and by one son, Anthony S., Jr. Their younger son, Wistar Morris, who served in the aviation service in the World War, was killed in France in 1918.

WILLIAM DYE MOUNT (1867–1934)

William Dye Mount, consulting mechanical and chemical engineer, of Lynchburg, Va., died suddenly at his home there on February 28, 1934, of angina pectoris. For many years he was managing director of all operations of the Mathieson Alkali Works, of Saltville, Va., and he was the inventor of a number of devices and processes for the chemical industry.

Mr. Mount was born on July 13, 1867, at Peruville, Tompkins County, N.Y., a son of William Everett and Lucretia Barbara (Giles) Mount. He attended the Groton Union School and received an M.E. degree from Cornell University in 1890. He secured experience in drafting with the Groton Bridge & Manufacturing Co. during summer vacations and for four years after his graduation he was on the faculty of Brown University, for three years as instructor in physics, then assistant professor of electrical engineering. He designed and laid out the equipment for the workshops at the University, and fitted up laboratories for electrical courses.

He entered the employ of the Mathieson Alkali Works in 1898. Concerning his work since that time Mr. Albert W. Smith, Cornell University, Class of 1878, stated, in an obituary published in the *Cornell Alumni News:*

"He was promoted, first to superintendent, and then to general manager, a position that he held with high efficiency for many years. He found this plant with antiquated apparatus and with an annual loss in operation; he designed and installed modern equipment, reorganized processes and methods, and brought the plant to a paying basis. In 1918 he left Saltville and opened an office in Lynchburg as consulting and designing engineer. He did important work in design and construction of apparatus for lime plants and pulp mills. In 1922 he designed and built an alkali plant in China which has been in successful operation ever since.

"In striving for the fulfillment of a long cherished dream, Mr. Mount acquired title to a bed of high-grade limestone near Natural Bridge, Va., and planned to construct a plant for production of pure lime that is so much in demand for modern chemical purposes; in addition he planned to extract and purify the carbon dioxide from the lime-kiln waste gases and solidify it as "dry ice" for purposes of refrigeration. This dream was based on sound financial and engineering judgment. But, alas, the depression came and checked development, and now death has taken the moving spirit and the dream has faded."

Mr. Mount held patents on processes for producing wood pulp and a continuous sulphate process for kraft pulp; power transmission and storage devices; methods and apparatus for handling foaming and frothing liquids; causticizing units; vertical and rotary kilns; process and apparatus for producing cyanide; flakers and filters; barrel heaters; apparatus for treating gases; automatic drains for compressors; and a car and vehicle loader. He was the author of "A Simple Method of Cleaning Gas Conduits," "Economic Waste and Inefficiency," "History of the Growth of an Important Industry," and other articles. He became a junior member of the A.S.M.E. in 1892 and a member five years later, and also belonged to the American Chemical Society, American Institute of Chemical Engineers, the Sigma Xi fraternity, and the Engineers' Club, New York.

Mr. Mount was twice married. His first wife, Miss Agatha Flanagan, of Seneca Falls, N.Y., whom he married in 1895, died in 1908, leaving three children. Later he married Miss Alice Martin, of Glade Spring, Va., who survives him, with their three children. There are two sons, Morris Blake and William D., Jr., and four daughters, Barbara, Eliza H., and Florence E. Mount, and Mary Agatha (Mrs. Ray) Mitchell. A brother, Joseph E. Mount, and three grandchildren also survive him.

WALTER JAMES MUNCASTER (1850-1934)

Walter James Muncaster, a member of the A.S.M.E. since 1887, died at his home in Cumberland, Md., on October 13, 1934.

Mr. Muncaster was born in Georgetown, D.C., on April 30, 1850, one of eight children born to Otho Zachariah and Harriet Elizabeth (Magruder) Muncaster. His father was a dealer and importer of hardware, and his mother the daughter of a lawyer of Rockville, Md. The family moved to Rockville shortly after the outbreak of the Civil War and there the children were brought up according to the strict standards of the Presbyterian church.

Walter, who had attended the school of Captain Thomas N. Conrad in Georgetown, completed his schooling at the Rockville Academy. The following interesting account of his boyhood days, written by Matthew Page Andrews, is found in the second volume of the Tercentenary History of Maryland, published by the S. J. Clarke Publishing Co., Baltimore, in 1925.

"After school hours and on holidays he did his share of the family work in the garden and at the woodpile, a familiar feature of the domestic landscape in those days. As a boy he exhibited an interest in mechanical devices that foreshadowed his achievements in mature life, but as his interest at that time found its outlet in dissecting clocks and locks his parents regarded it more as an evidence of original sin than inventive genius. The great event of his boyhood was, of course, the Civil War, and with all his boyish love of things military he longed to be a real soldier. At one time in the course of the war, in company with some playmates, he did run away to enlist, in spite of the fact that he was too young to be taken into the army in any capacity. He was found in the course of a few days and sent home to the prosaic tasks of the schoolroom."

At the age of seventeen Mr. Muncaster went to New York, where he served an apprenticeship as a machinist at the Novelty Iron Works. At the same time he attended evening classes in engineering at Cooper Institute, securing an M.E. degree in 1870. After completing his apprenticeship he worked successively as a machinist for Duvall & Pierce, Georgetown, at the U.S. Navy Yard, Washington, and in the shops of Poole & Hunt, Baltimore. Of his subsequent experience Mr. Andrews writes:

"In 1874 he came to Cumberland as a machinist in the Beall Foundry, now known as the McKaig Foundry & Machine Shop, where he was holding an executive position when he left to assist in the organization of the Cumberland Steel Company. This firm was established in 1892, and since then Mr. Muncaster has been vice-president and general manager. The product is the highest grade of steel shafting, which is made possible by some twenty-two inventions of Mr. Muncaster's for the perfection of the machinery used in the plant."

Mr. Muncaster had not been active in business since 1921. The complete list of patents granted to him between 1875 and 1913 contains 27 items. They include machines for straightening and finishing shafting, metal bending and straightening machines, grinding machines for glassware, measuring instruments, lathes for metal working and turning shafting, machines for boring and turning pulleys and for boring cylinders, a crane for loading box cars, a pulley chuck, raising and lowering screw-propellers, tool and work holders, apparatus for applying couplings to and removing them from shafting, and other shafting machinery, as well as early patents on a portable steam engine, gridiron, and water-tube steam boiler.

Mr. Muncaster was twice married. His first wife, Miss Anna R. Lewis, of Georgetown, whom he married in 1874, died two years later and their only child, Rosalie, died in 1902. He is survived by his widow, Mary I. (Spear) Muncaster, whom he married in 1904, and by their daughter, Margery Ivolue Muncaster.

JOHN CLEMENT NAEGELEY (1844–1935)

John Clement Naegeley was born on March 30, 1844, in the town of Ulm, Württemberg, Germany. He came with his parents to Hollidaysburg, near Altoona, Pa., in 1855, and lived there with them until 1864 when he returned to Germany for a visit.

His education began at the public schools at Ulm, and, after his parents came to Pennsylvania, was continued at schools and under a private tutor at Altoona and later at St. Vincents College, Latrobe, Pa.

After his return from Germany he held positions in drafting rooms and shops, designing and constructing machinery, locomotives, buildings, and bridges for the following companies: 1867 to 1870 with the Altoona & Great Western R.R. at Kent, Ohio; 1870–1871 with the Cleveland Bridge & Car Works, Cleveland, Ohio; 1871 to 1874 with the Porter Locomotive Works, Pittsburgh, Pa.; 1874 to 1877 with the New York Central Railroad at Albany, N.Y.; 1877 to 1878 with the New York, Boston & Albany R.R. at Boston, Mass.; 1878 to 1879 with the Cleveland & Akron R.R. at Cleveland, Ohio; 1879 to 1885 with the Keystone Bridge Works at Pittsburgh, Pa.

In 1885 he started in business for himself, presumably in the inspection of iron and steel work at the mills and shops, but no record of his work during this period is now available until 1901, when he was employed as chief inspector at the mills and shops of the structural steel work for the Atlantic Avenue Improvement of the Long Island Railroad in Brooklyn, N.Y. This was a grade separation undertaking of considerable magnitude involving the rolling and fabrication of some 12,000 tons of structural steel work, including the tonnage in two heavy railroad viaducts with a combined length of about $2^1/_2$ miles in the streets of Brooklyn. This work kept Mr. Naegeley employed from 1901 to 1904.

Late in 1904 he accepted employment under Charles M. Jacobs, consulting engineer, first inspecting structural materials for the Hudson & Manhattan R.R. tunnels and later as chief metal inspector, at the foundries and shops, of the cast-iron lining and other structural metal for the Pennsylvania Railroad Tunnels, under the Hudson River, and continued in that position until the work was finished.

From about 1912 to 1916 he was engineer of inspection, under Gustav Lindenthal, chief engineer, in charge of steel inspection for the Hell Gate bridge and approaches.

Later he was inspector of steel work for the Delaware River Bridge between Philadelphia, Pa., and Camden, N.J., under Ralph Modjeski, chief engineer, and while the dates and details of this latter engagement are not now available, Mr. Modjeski writes under date of July 10, 1935: "Mr. Naegeley performed a great deal of steel inspection for me both at the mills and at the shops in a most thorough and conscientious manner."

Another prominent engineer who was a personal friend of Mr. Naegeley and had great confidence in his reliability and thoroughness as a steel inspector, was the late Charles C. Schneider, past-president of the American Society of Civil Engineers, and for many years chief engineer of the American Bridge Company, who recommended his services to others for much important work.

The words "thorough" and "conscientious" come naturally to the mind of any one who knew John Naegeley and his work; and that characterization could be very truthfully applied to everything he did. He was thoroughly loyal to those by whom he was employed, and uncompromising in his insistence on compliance with both the

letter and the spirit of the specifications but, at the same time, fair and just in dealing with those honestly trying to do good work.

Mr. Naegeley became a member of the A.S.M.E. in 1901.

He died on April 6, 1935, at the unusual age of 91 years. The last years of his life were spent at the home of his nephew, J. F. Varcoe, at Maplewood, N.J.—[Memorial prepared by JAMES B. FRENCH, Consulting Engineer, New York, N.Y., with the cooperation of J. F. VARCOE, Mr. Naegeley's nephew, and of MESSRS. J. VIPOND DAVIES, O. H. AMMANN, and RALPH MODJESKI, all engineers who were familiar with Mr. Naegeley's work at various times.]

JOHN K. OLSEN (1887–1935)

John K. Olsen, for many years chief draftsman of the Stewart Warner Corporation, Chicago, Ill., died at his home in that city on October 24, 1935, after a prolonged illness.

Mr. Olsen, a native of Norway, was born at Horten on October 16, 1887, the son of Nitinius and Marie Olsen. He held a scholarship at the Horten Technical College, from which he was graduated with an M.E. degree in December, 1906. He came to the United States soon afterward and was employed until the end of 1909 by the Crane Company, Chicago, as machinist and draftsman. He then entered the Hawthorne works of the Western Electric Company, where he worked as draftsman in connection with mechanical and electrical apparatus for four and one-half years. After another year in checking apparatus drawings he was made head of a subsection responsible for important work in the apparatus-manufacturing drafting room.

Mr. Olsen became associated with the Stewart Warner Speedometer Corporation in June, 1918, in the position of designer and checker, and was made chief draftsman a few months later. He had charge of engineering design and production specifications. He held patents on many of his own designs in automobile accessories and helped to develop a large number of items produced by the company. He also compiled several hundred standard specifications used by the company. Unable to continue with his work for the Stewart Warner Corporation during the last year of his life, Mr. Olsen had done some temporary work for the Zerk Corporation and Florence Stove Company, Kankakee, Ill.

His long experience in drafting and standardization work led to his appointment as representative of the A.S.M.E. on the Sectional Committee on Drawings and Drafting Room Practice and he served on two of this group's subcommittees—Fine Work and Graphical Symbols on Drawings. He became an associate-member of the Society in 1921 and a member four years later.

Mr. Olsen was the author of "Production Design," published in 1928, contributed to the "Cost and Production Handbook," and wrote numerous articles for the technical press. He became a naturalized citizen of the United States in 1916.

Mr. Olsen is survived by his widow, Marie (Haugen) Olsen, and by one son, Canute Roald Olsen.

FRANCIS JOSEPH OTTIS (1871–1935)

Francis Joseph Ottis, founder and president of the Northern Malleable Iron Company, St. Paul, Minn., died in that city on January 5, 1935. He was born at Central Mine, in Northern Michigan, on September 18, 1871, attended public school in Nebraska, and was graduated from Creighton University, at Omaha, with an A.B. degree, and from Harvard University in 1896 with an LL.B. degree.

After practicing law in New York, N.Y., for two years he went to St. Paul where he was connected with the American Grass Twine Company as general manager for two years and with the Minnie Harvester Company in the same capacity until he organized the Northern Malleable Iron Company in 1902.

When Mr. Ottis became a member of the A.S.M.E. in 1923 he wrote of his experience:

"I have had to do, in a supervisory capacity, with engineering problems including the clearing and ditching of thousands of acres of marshlands of Minnesota, building of manufacturing plants, the building and operation of intricate machinery for the manufacture of grass twine, grass carpets, and flax twine; later, the supervision of inventions which were patented for the manufacture of grass twine and flax twine and the manufacture of harvesting machinery involving very many patents.

"In the manufacture of malleable castings I have been active in raising the quality of malleable iron and in introducing new methods and devices in the industry which in cooperation with others has resulted in raising specifications for malleable castings by the American Society for Testing Materials, from the old standard, 38,000 lb tensile, $2^1/_2$ per cent elongation, 2 in., to the present standard of 50,000 lb tensile and 10 per cent elongation, 2 in."

During the World War period he was a special representative of the malleable iron castings industry to handle matters coming before the War Industries Board, particularly those dealing with production. He also directed the preparation of price classification schedules.

Mr. Ottis was greatly interested in educational and philanthropic enterprises and also in fine etchings. He is survived by his widow, Laura D. (Cook) Ottis, whom he married in 1891, and by three married daughters, Elizabeth Clark, Clara Louise Harris, and Laura Frances Bradshaw.

JOSEPH YALE PARCE (1870–1935)

Joseph Yale Parce, who died at his home in Denver, Colo., on April 18, 1935, was born at Fairport, N.Y., on July 13, 1870, son of Joseph Yale and Lucy (Mead) Parce. His childhood, however, was spent at DeLand, Fla., and he was graduated from the Academy there in 1889. He then entered the Massachusetts Institute of Technology, from which he received an S.B. degree in mechanical engineering in 1893.

During the summers of 1891 and 1892 Mr. Parce had been employed by the American Tool & Machine Co., of Boston, and he continued with the company until June, 1895, being engaged in design, construction, and installation of various kinds of mill and factory equipment, including power-plant and transmission machinery. He went to Denver in July, 1895, to design and install equipment for the shops and power plant in the Manual Training High School, and to develop courses in shopwork. He continued to serve the public school system of Denver until ill health forced him to retire in August, 1934.

Mr. Parce had taught mechanical drawing, machine shop work, and automobile repairing, in evening and summer schools. He had served as director of industrial arts (including vocational shopwork) at the Manual Training High School since November, 1919. He was also in charge of the Y.M.C.A. Automobile School in Denver during the summer of 1910, and was consulted frequently by various industries of Denver on establishing equipment in their plants, and on design and patent work.

Mr. Parce became an associate-member of the A.S.M.E. in 1915. Although he did not participate in their programs in any definite way, he kept up faithful membership in several state and national vocational and educational associations. He was active in church and Masonic work, having served several terms as elder and clerk of the session of the Montview Presbyterian Church of Denver, and belonging to the Knights Templar and Blue Lodge. He had a workshop of his own at home and spent much time there and, for some years, in gardening. He is survived by his widow, Inez (Taggart) Parce, and by two sons, Joseph Yale, Jr., and Earl T. Parce.

HARRY DE BERKELEY PARSONS (1862–1935)

Harry de Berkeley Parsons was born on January 6, 1862, in New York, N.Y., the son of William Barclay Parsons and Eliza Glass (Livingston) Parsons. He was descended from distinguished Colonial stock, long identified with the development of New York city. Among his direct ancestors were Robert Livingston, First Lord of the Manor, who came to this country from Scotland in 1673; Cadwallader Colden, Lieutenant-Governor of the Province of New York in 1760; and Colonel Thomas Barclay, who was appointed the British Consul General for New York, after the American Revolution. A brother who distinguished himself as an engineer and a soldier of the United States was the late William Barclay Parsons, who was chief engineer of the Rapid Transit Railroad Commission during the construction of the first Rapid Transit Subway for New York.

In 1870, Mr. Parsons went to Europe with his family and during the following four years studied under private tutors, while traveling in France, Germany, and Italy. He matriculated at Columbia College, in New York, in 1878, from which he was graduated with the degree of bachelor of science in 1882. He then entered Stevens Institute of Technology, in Hoboken, N.J., from which he was graduated in 1884 with the degree of mechanical engineer. In 1926, Stevens conferred on him the honorary degree of doctor of engineering.

In 1885, Mr. Parsons established an office in New York city as a consulting engineer and maintained it there until his death. His field of practice was an unusually diversified one. To recite all the activities of his fifty years of experience as a consulting engineer would make too formidable a list to publish herein, but one who has studied them is impressed with their number, and with the versatility of mind of this quiet, conscientious man. Among the earlier projects with which he was connected was the construction of the Fort Worth and Rio Grande Railway, in Texas, including the erection of a bridge over the Brazos River, and, in 1889, the completion of the water supply for Stevens Point, Wis. In 1893, he was consulting engineer for the Nicaragua Canal Construction Company and some of its related organizations.

From 1892 to 1907, he was professor of steam engineering at Rensselaer Polytechnic Institute, Troy, N.Y., and from the latter date until his death was its emeritus professor of practical engineering.

In 1898 he was consulted on the adequacy of the foundations for the Protestant Episcopal Cathedral of St. John the Divine in New York city. At a later date, he designed and installed the heating system for the Cathedral, and also had to do with the design of some of the appurtenant structures.

In the period, 1901 to 1903, Mr. Parsons designed the Spiers Falls Dam and Power House, on the Hudson River, at Spiers Falls, N.Y.—1570 ft long, with a maximum height of 154 ft—and, in the period 1921 to 1923, the Sherman Island Dam on the same river, near Glens Falls, N.Y., for the International Paper Company. He also acted as consulting engineer while these projects were being constructed.

Between 1898 and 1914 he was a member of the New York State Voting Machine Commission, and was chairman of the Mayor's Committee to Report on Street Cleaning and Waste Disposal, in New York city, in 1906 and 1907. From 1908 to 1914, he served as a member of the Metropolitan Sewerage Commission which prepared a monumental report on the Storm and Sanitary Drainage of New York city; and from 1898 to the time of his death Mr. Parsons was a consulting engineer for the New York Zoological Society.

His activities included the designing of water and steam power plants for mills; roof trusses, heating plants, and other features of church buildings; water-supply and sewerage projects for cities; many designs for hydroelectric developments, and numerous appraisals of industrial projects, water powers, and railroads in many states. He designed the foundations and underpinning for many structures for construction contractors, in connection with the building of the Rapid Transit Subway in New York city; and in 1918 and 1919, as district appraisal officer at Detroit, Mich., Mr. Parsons represented the United States Government in the settlement of World War contracts connected with the Air Service and aircraft production. He acted as consulting engineer for the Cramp Ship and Engine Building Company on reports and appraisals; for the Pressed Steel Car Company; for The Consolidated Gas Company of New York; numerous paper companies; the Seaboard Air Line; and the New Hampshire Traction Company. He was consulting engineer for the Department of Street Cleaning of New York city through five administrations, during which he designed and built two rubbish destructor plants and prepared the design for a third for the building of which the city did not appropriate the necessary funds. For a time, 1905–1906, he was the consulting engineer for the Fire Department and for the Department of Docks and Ferries, of New York city, and designed and constructed a number of fireboats and ferryboats for these departments.

For many years Mr. Parsons was the consulting engineer for the Corporation of Trinity Church, the Rhinelander Estates, the Corn Exchange Bank, the Mechanics and Metals Bank, and the Bush Terminal, all of New York city. The Delaware and Hudson Company retained him to report on and appraise some of its electric railway properties, the construction cost of which amounted to more than $14,000,000. The City of New York employed him to appraise the damages to the Ulster and Delaware Railroad by the flooding of the Ashokan Reservoir, for the properties taken for the large Chelsea Dock improvement, on the Hudson River front, and for the new County Court House—the condemnation for all of which amounted to many millions of dollars.

He regarded as his most important accomplishments the Spiers Falls and the Sherman Island Dams, the study of tidal flows in the Harbor of New York for the Metropolitan Sewerage Commission, and the appraisals at the end of the World War for the U.S. Army Air Service plants, at Detroit.

Mr. Parsons was a liberal contributor to the publications of the engineering and scientific bodies to which he belonged, both in original papers and in discussions of those of other authors. He was awarded the Thomas Fitch Rowland Prize by the American Society of Civil Engineers in 1925 for his paper on "Sherman Island Dam and Power-House," and, in 1930, was given the J. James R. Croes Medal of that society for his paper on "Hydrostatic Uplift in Pervious Soils." Mr. Parsons was also the author of "Disposal of Municipal Refuse and Rubbish Incineration," "Tidal Phenomena of the Harbor of New York," and "Some Soil Pressure Tests," published in the Transactions of the A.S.C.E. Papers presented before the A.S.M.E. and published in its Transactions included "Standard Cross-Sections," "Comparison of Rules for Calculating the Strength of Steam Boilers," "The Influence of Sugar Upon Cement," and "The Displacements and the Area-Curves of Fish." His book entitled "Steam Boilers, Their Theory and Design" (1917), went through five editions.

Mr. Parsons became a junior member of the A.S.M.E. in 1885 and a member two years later. He served on the Publications Committee from 1890 to 1902. He was elected to the Meetings Committee in 1903, and also served on this committee from 1909 to 1914. He acted as chairman of the representatives of the Society on the Joint Committee on Fire Proofing Tests, whose report was published in Volume 18 (1897) of Transactions. He was also chairman of the Committee on Standard Cross-Sections and Symbols, whose report was published in Volume 36 (1914) of Transactions. He was elected a manager of the Society for the term 1915–1918.

Mr. Parsons had been a member of the American Society of Civil Engineers since 1897. He also belonged to the Society of Naval Architects and Marine Engineers and the American Institute of Consulting Engineers, of which he was the president in 1926. He was chairman of the Professional Engineers Committee on Unemployment, which represented the four Founder Societies, in 1931–1932, and was on its Advisory Board in 1932 and 1933. He was a member of the Delta Psi Fraternity and served as president of the Alumni Association of Stevens Institute in 1895 and 1896 and as Alumnus Trustee from 1896 to 1899.

In addition, Mr. Parsons was active in civic affairs, and was a member of the New York Zoological Society and of its Board of Trustees, the Chamber of Commerce of the State of New York, the Merchants Association of New York, the Metropolitan Museum of Art, New York, and was a trustee of the United Hospital of Portchester, N.Y., and chairman of its Endowment Fund.

He belonged to the Protestant Episcopal Church and was a member of the vestry of the Church of the Incarnation, in New York city, where his funeral services were held, and of Christ Church, at Rye, N.Y., where his beautiful summer home is located. He belonged to the following clubs in New York city: Union, Engineers, New York Yacht, and the Downtown Association, and to the American Yacht Club, and the Apawamis Club, in Rye, N.Y.

Mr. Parsons was a keen yachtsman and as a young man gained much experience at the helms of his own boats and of his father's yacht. He was much interested in this clean sport, and for many years subsequent to 1895 served as chairman of the Race Committee of the American Yacht Club. From 1904 to 1922, he was also a member of the Racing Committee of the New York Yacht Club, being chairman from 1907 to 1922. The *New York Times* in an article published concerning him on January 27, 1935, quotes this statement of his made in 1917:

"Yachting has a peculiar charm which is difficult to describe. I fancy it comes from the feeling of freedom and from the intimate companionship which is so pleasant. Men who are fond of the sea are usually fearless, frank and good sportsmen, qualities which make for staunch friendships and the pleasantest associations."

One feels how inadequate is this recital of Mr. Parsons' deeds as a measure of what he accomplished during his life. When meeting him, one always felt that he was with a man who, by instinct and training, was a finished engineer. Always courteous, his quiet refinement of manner, his sincerity of purpose, his modesty, and his grounding in the fundamentals of his profession distinctly impressed one. A cultured gentleman possessed of broad knowledge, his life was unusually full and useful. In spite of his modesty time will appraise the real value of his contributions to his profession.

An editorial in *Mechanical Engineering*, March, 1935, says of him:

"To him the profession he served had more than ordinary significance. He held its members in respect and affection, and to its organized activities—the engineering societies—he gave liberally, at times of his experience in the technical papers he wrote, at times of his able advice and services, at times of his money. His warm sympathies were deeply touched by the plight of hundreds of brother engineers who suffered from the depression, and though not robust in health himself, he found time actively to aid in the organized relief of the Professional Engineers' Committee on Unemployment. To it he contributed liberally. The appeal to buy bonds of The American Society of Mechanical Engineers found him responsive. The United Engineering Trustees, Inc., he remembered in his will. Thus in many and varied ways he identified himself with his profession."

He was married on December 16, 1890, to Frances Thompson Walker, of New York city, who died in 1917. He is survived by a son, Livingston Parsons, and a daughter, Katherine de B. Parsons, and three grandchildren.

Mr. Parsons died at his home in New York on January 26, 1935.

[From a memoir prepared by Robert Ridgway and Lynne J. Bevan for the Transactions of the American Society of Civil Engineers.]

EDGAR PIERCY (1856–1936)

Edgar Piercy, who died at Hamilton, Ontario, Canada, on January 20, 1936, was born on April 6, 1856, at Brighton, Sussex, England, son of Christopher and Harriet (Trusler) Piercy. Having completed his early education in common schools and at the Croydon Boys

School he was apprenticed to John Oldfield, architect and surveyor, of Croyden, Surrey, England. During his apprenticeship and afterward he studied mechanical engineering, and later he attended high school in Hamilton.

Mr. Piercy's first engineering work in Canada was for the Consolidated Purifier Company, of Toronto, builders of flour mills and flour-mill machinery, for whom he was draftsman for a number of years, and subsequently he was in charge of drafting for a company engaged in a similar line of work at Stratford, Canada. This company, the Geo. T. Smith M. P. Co., transferred him to its head office at Jackson, Mich., in 1886, and from there he went to Camden, N.J., to become draftsman for the Camden Iron Works, of whose hydraulic traveling crane department he was later put in charge. He was also engaged in work for Poole & Hunt, of Baltimore, and R. D. Wood & Co., of Philadelphia, during his early years in the East.

About 1893 he became superintendent of Reeves Bros., at Niles, Ohio, and was located there and at Alliance, Ohio, until 1895. He then went to Johnstown, Pa., where he was employed by the Cambria Steel Company as assistant engineering superintendent for about four years. His next position was that of manager and superintendent of the Milwaukee (Wis.) Boiler Company, manufacturers of marine and stationary boilers, tanks, etc. In 1901 and the early part of 1902 he was connected with the Carpenter Steel Company in Reading, Pa.

With this varied background he entered the service of the United States Steel Corporation at Lorain, Ohio, in May, 1902, as blast furnace engineer. He held this position until his transfer in June, 1923, to the Gary, Indiana, Works of the National Tube Company where new pipe mills were being erected. As assistant chief mechanical engineer he aided in the construction and operation of this unit until he retired on a pension in April, 1931. In 1927 he was presented with a 25-year medal by the United States Steel Corporation in recognition of his service to the industry.

Mr. Piercy had been a member of the A.S.M.E. since 1881. His pastimes were drawing, painting, woodworking, and the illumination of presentation addresses. His wife, Margaret (Cusack) Piercy, died in 1927. They had been married since 1885.

WILBER OSBORNE PLATT (1860–1934)

Wilber Osborne Platt, president of the Joseph Reid Gas Engine Company, Oil City, Pa., died in that city on April 18, 1934. He had been a member of the A.S.M.E. since 1902 and was keenly interested in the work of the Society.

Mr. Platt was born on January 4, 1860, in Clarion County, Pa., son of Hugh and Mary A. (Echelbarger) Platt. He attended the Soldiers Orphans' School of Pennsylvania and later took courses through the International Correspondence Schools.

At the age of sixteen he was apprenticed to W. J. Innis & Co., in Oil City, to learn the machinists' trade. After two years' training he entered the employ of Joseph Reid, for whom he worked until 1882. During the six years following he was with Harman Gibbs & Co. (which later became the Ajax Manufacturing Company), engaged in repair work, part of the time as foreman of a repair crew. He then returned to the employ of Mr. Reid in the capacity of general foreman, his work including the design of fixtures and full charge of a general jobbing shop.

When the Joseph Reid Gas Engine Company was incorporated in 1899 he was made superintendent, responsible for all designs and manufacturing methods. He became a vice-president of the company after a few years and continued to serve in that capacity, as one of the directors, and as superintendent until the death of Mr. Reid in 1917, when he was elected president of the company. He was also president of the Reid Land & Development Co. and vice-president of the Frick-Reid Supply Corporation.

Mr. Platt held patents on horizontal power-transmission wheels; vaporizers and igniters for internal-combustion engines; grinding machines; bearings for oil-well jacks and swing levers; governing valves for gas engines; governors with safety-trigger arrangement; internal-combustion engines; and gas-purifying apparatus for cleaning natural gas.

He was a frequent contributor to the technical press, particularly to the *American Machinist*, for which he wrote a series of articles on "Echoes from the Oil Country," under the pen name of W. Osborne. These were begun about 1900 and continued irregularly for many years. They were exceptionally practical letters and attracted considerable attention. He was a student of biology and astronomy, fond of outdoor life, and devoted to the interests of his employees. He was a member of the Odd Fellows.

Surviving him are his wife, Lucinda A. (Messenger) Platt, and six children, Mrs. Annie L. Brakeman, Mrs. Rose A. Ramsey, Mrs. Mary Lou Bellen, and Olive M., Hugh A., and J. Reid Platt.

JAMES M. PONISOVSKY (1901–1935)

James M. Ponisovsky, whose death occurred in Brussels, Belgium, on December 31, 1935, was born in Moscow, Russia, on December 26, 1901. He was the son of Matthew and Amalia Wilenkin Ponisovsky of Tsarskoë Selo. After attending a private school in Moscow he was obliged to leave the country with his family owing to the Bolshevik Revolution. On reaching London in January, 1919, he was tutored at Payen Payne for the City and Guild's Engineering College of the University of London, from which he was graduated with a B.Sc. degree in 1923.

Mr. Ponisovsky was employed by John Hetherington & Sons Limited, at the Vulcan Works, Manchester, England, as an apprentice during the remainder of 1923, fitting and erecting cotton-spinning machinery, and during the next year was a special apprentice on the erection of textile machinery with Mather & Platt, Ltd., at the Park Works, Manchester.

After completing his apprenticeship Mr. Ponisovsky took up the study of production of tinplate for The Melingriffith Company, Ltd., at Whitchurch, near Cardiff, South Wales. He continued with this company until May, 1926, serving during the latter part of the time as liaison official between the office and the production department.

Mr. Ponisovsky then came to the United States and in July, 1926, became research assistant for the Baltimore Copper Smelting & Rolling Co., Baltimore, Md., a subsidiary of the American Smelting & Refining Co. Since the first of January, 1930, he had been connected with Copper Exporters, Inc., of New York, N.Y., representing the copper industry in Europe, investigating complaints and giving technical advice on the fabrication and use of copper.

Mr. Ponisovsky became an associate-member of the A.S.M.E. in 1929. He also belonged to the American Institute of Mining and Metallurgical Engineers and the Institution of Mechanical Engineers. He is survived by his father, Matthew Ponisovsky, and by two sisters and a brother, all residing in Paris, France.

JOHN FREDERICK POOL (1863–1935)

John Frederick Pool, a native of Cornwall, England, died at his home at Falmouth on August 29, 1935. He was born at Perran-ar-Worthal on February 11, 1863, son of Francis and Anne Maria Sara Pool. He attended a private school at Devoran and served a five-year apprenticeship with Cox & Co., engineers and shipbuilders of Falmouth.

After several years' experience as an erector of marine engines and boilers in shops and on board steamships, Mr. Pool came to the United States in 1888 and was first employed here as draftsman for the Spreckels Sugar Refinery, of Philadelphia, Pa. He rose through the positions of chief draftsman, first assistant engineer, and chief engineer of that company to the post of superintendent of the American Sugar Refining Company, at its Brooklyn Refinery, which he held from 1909 until his retirement in April, 1916. His invention of an improvement on a mechanical stoker was purchased by the company.

Since his retirement Mr. Pool had devoted much of his leisure to gardening and in 1929 won a gold medal for his exhibit at the Falmouth Flower Show. He is survived by a niece, Mrs. A. Muriel Evans, of Falmouth. His wife, Mary E. (White) Pool, whom he married in 1893, died in 1914.

Mr. Pool had been a member of the A.S.M.E. since 1914.

MINOTT E. PORTER (1869–1936)

Minott E. Porter, principal examiner, United States Patent Office, died in Washington, D.C., on February 26, 1936. He was born at Hinckley, Medina County, Ohio, on July 11, 1869, son of Henry T. and Lydia Porter. His boyhood was spent on a farm. During the school year of 1888-1889 he attended Hiram College. Deciding to take an engineering course, he entered the University of Michigan in 1889 and was graduated four years later, receiving the degree of B.S. in civil engineering. Continuing his studies, he secured his C.E. degree from the same university in 1897.

Prior to his graduation Mr. Porter secured engineering work during the summer of 1892 as one of the inspectors of masonry on an 800-foot lock being constructed at Sault Ste. Marie, Mich. Owing to a period of depression he was unable to secure any other special work until the fall of 1894, when he was appointed a computer in the United States Hydrographic Office of the Navy Department, Washington, D.C. In the early part of 1897 he was transferred to the United States Naval Observatory at Washington.

Mr. Porter began work in the Patent Office in December, 1901, as fourth assistant examiner. He was promoted through the several grades and early in 1914 was appointed law examiner. During the

latter part of this period he examined classes of patents dealing with rock drills, mining machinery, and stoneworking machinery. From 1902 to 1906 he studied law at George Washington University, receiving the degree of master of patent law. As law examiner, 1914–1915, he was a special assistant to the Commissioner of Patents, devoting his time to law matters and to the supervision of declarations of interferences involving inventions of all kinds.

When Mr. Porter was made principal examiner in November, 1915, he was assigned to Division 18, which handles classes of patents dealing with steam engines, speed-responsive devices, boilers, and power-plant equipment. His work involved the analysis of mechanism in the foregoing classes to determine its workability, novelty, and patentability. He was a member of the bar of both the Supreme Court and the Court of Appeals of the District of Columbia.

Mr. Porter became a member of the A.S.M.E. in 1918 and was a past-chairman of the Washington Section of the Society, which he helped to organize.

Mr. Porter married Alice M. Palmer, of West Richfield, Ohio, in 1894, and is survived by her and their son, Clarence H. Porter, of Boston, Mass.

HARRY CADWALLADER RAYNES (1871–1933)

Harry Cadwallader Raynes, who died on April 18, 1933, was born at Lowell, Mass., on August 18, 1871. He supplemented his high-school course with special work in mechanical engineering and economics.

After some preliminary experience Mr. Raynes took a position as mill engineer in Lowell in 1897. In 1900 he went to Dover, N.H., as construction engineer for the Cocheco Manufacturing Company, with which he was connected for three years. In 1904 he was associated in engineering work with Stephen T. Williams, of New York, then became supervising engineer and works manager for the Eaton Hurlbut Paper Company, Pittsfield, Mass. He engaged in consulting work in Boston from 1907 to 1909 and spent the following year with the Emerson Company, New York, in the capacity of directing engineer. From then until 1914 he was consulting engineer and mechanical superintendent of the paper mill of F. W. Bird & Son, East Walpole, Mass.

Since 1914 Mr. Raynes's work had been chiefly in the consulting field, his office being in New York the greater part of the time. During the War years he was general manager of the Atlantic Shipbuilding Corporation at Portsmouth, N.H., and in 1921 went to the Minn. & Ontario Paper Co. as consulting engineer. In 1924 he was consulting engineer at the Johnson & Canden Co. in New York, and for approximately three years, beginning in June, 1926, he was vice-president and general manager of the New England Division of the Atlantic Gypsum Products Company, of Boston.

Mr. Raynes became a member of the A.S.M.E. in 1913.

WILLIAM L. REID (1864–1935)

William L. Reid, vice-president, in charge of manufacturing, of the Lima Locomotive Works, Inc., Lima, Ohio, died on March 9, 1935. He had been a member of the A.S.M.E. since 1930.

Mr. Reid was born in Paterson, N.J., on October 3, 1864, the son of John I. and Agnes (Rankin) Reid.

Mr. Reid's father, John I. Reid, was born in Scotland and came to this country while still in his teens. He served as an infantry private in the Union Army during the Civil War. Following his discharge, he spent the remainder of his life in the locomotive industry in Paterson. Without the benefit of any education other than that acquired by his own reading, he worked his way up to the position of general foreman at the Rogers Locomotive Works. He furnished three sons to the locomotive industry—James A. Reid, retired by the American Locomotive Company, Charles L. Reid, draftsman of the American Locomotive Company, and William L. Reid, the subject of this obituary.

William L. Reid attended the public grade and high school until in 1880 when he began a drawing-room apprenticeship with the Rogers Locomotive Works in his native city. After completing a four-year drawing-room apprenticeship, he served his time as a machinist apprentice with the same company.

Completing that apprenticeship he served as a machinist on the New York Manhattan Elevated Railroad until 1887, at which time he was appointed a foreman of the erecting shop of the Rogers Locomotive Works in Paterson. In 1899 he was appointed assistant superintendent of the same company. In 1902 he was appointed assistant superintendent of the Brooks Locomotive Works, which position he had held only a few months when he was made superintendent. In 1903 he was made superintendent of the Schenectady Works of the American Locomotive Company. His ability was so exceptional that later in the same year he was made manager of the works. He served in that capacity until 1907, when he was appointed general works manager.

In 1917 he was appointed general superintendent of the Baldwin Locomotive Works, a position that he held for two years. During that period he made a noteworthy record not only in connection with locomotive construction but in the fabrication of various war materials.

In 1918 he was elected vice-president in charge of manufacturing of the Lima Locomotive Works, Inc., a position which he held until the time of his death. With that company he made an outstanding record for quality production, inventive capacity, and administrative ability.

Mr. Reid was a valuable citizen in the communities he served. In Schenectady, N.Y., he was a member of the Board of Education and the Park Commission. He took a very great interest in civic affairs and at all times did much to inspire both young and older generations in the importance of constructive citizenship. He was fearless in advocating a principle that he believed to be right or just. Although exacting in discipline, he tempered his acts with impartial justice.

In 1885 he married Elizabeth Hammond, who died in 1927. Surviving Mr. and Mrs. Reid are a number of distinguished children, Dr. Ralph D. Reid, Wm. H. Reid, John I. Reid, Leslie Reid, and Mrs. R. H. Pew.

Mr. Reid was ever a keen and devoted student. He took exceptional interest in astronomy and in American history, particularly that period dealing with the Civil War.

In his varied and remarkable career Mr. Reid made substantial contributions to the sum of locomotive construction knowledge. He was an acknowledged authority in such matters.

Able in his profession, devoted to duty, and imbued with the dignity of self respect, he was an outstanding engineer and citizen, and a fine type of the true American gentleman.—[Memorial prepared by W. H. Winterrowd, Chicago, Ill. Mem. A.S.M.E.]

CHARLES RUSSELL RICHARDS (1865–1936)

Charles Russell Richards, engineering educator and leader in the fields of industrial education and industrial museums, died at the Doctor's Hospital in New York, N.Y., on February 21, 1936.

Professor Richards was born at Boston, June 30, 1865, the son of Charles C. and Josephine (Gleason) Richards. He was educated in the High School at Roxbury, and at the Massachusetts Institute of Technology, where he was graduated in 1885, from the mechanical engineering course. After graduation he was chief draftsman and assistant superintendent at the Whittier Machine Company of Boston, until 1887. He left this position to take charge of the shops of the Industrial Education Association of New York, and remained there for about a year. From 1888 to 1898 he was director of the Department of Science and Technology of Pratt Institute in Brooklyn. From 1898 to 1908 he was director of the Department of Industrial Art at Teachers College, Columbia University. From 1908 to 1923 he was director of Cooper Union and chief administrator of the entire institution. During the World War he organized here special classes in war industry work, which were very successful. His wide knowledge of economic history, of all phases of industrial life, and of the trade routes of the world, made the lectures which he gave on these subjects as part of the engineering curricula, of great value. His familiarity with many phases of art fitted him admirably for directing the work of Cooper Union in its objective of advancement of science and arts. From 1923 to 1926 Professor Richards was director of the American Association of Museums. From 1926 to 1930 he was in charge of the Division of Industrial Arts of the General Education Board. In 1930 he became executive vice-president and director of the New York Museum of Science and Industry, and held this position until the failure of his health in 1935.

This Museum, started as the Museum of Peaceful Arts through the bequest of Henry R. Towne, had made a small beginning and, for a number of years, had been trying to crystallize the problem before it. Under Professor Richards' leadership a comprehensive plan was outlined, the Museum was moved to larger quarters, and an exhibit opened which covered the fields of food, housing, textiles, transportation by highway, rail, water, and air, communication, machine tools, power, and electrotechnology. It included also an auditorium where each day moving pictures told the story of various industries and phases of arts and manufactures. The Museum has outgrown these quarters and was moved shortly before his death to Rockefeller Center, where it is going forward upon the foundations laid down by him.

Professor Richards joined the A.S.M.E. in 1890. He was president of the National Society for the Promotion of Industrial Educa-

tion from 1909 to 1913, a trustee of the Children's Aid Society of New York from 1904 to 1916, a member of the Corporation of the Massachusetts Institute of Technology from 1909 to 1914, and a member of the Century, Art-in-Trades, and also of the Technology Club, of which he was at one time president.

In addition to the above activities he rendered many other valuable services. He was on the Advisory Commission on Industrial Education for the New York State Board of Regents, was chairman of the Commission to visit and report on the Paris Exposition, and he organized the National Society for Vocational Organization. He was author of "Arts in Industry," "The Industrial Museum," and "Industrial Art and the Museum." In April, 1926, he received the Cross of the Legion of Honor of France, and in 1935 was awarded the Michael Friedsam Medal by the Architectural League of New York. Only a few hours before his death a portrait bas relief of him was dedicated at the entrance of the Engineering Building of Pratt Institute.

Professor Richards was married in 1917 to Hilda Muhlhauser, and in 1926 to Mildred Batchelder, who survives him.

Professor Richards united a wide knowledge of modern industrial production and an unerring taste, a combination which made him a leader in guiding modern industrial activities along lines combining utility and beauty. His ranging mind, enriched by travel and wide reading, his great personal charm, and his generous and loyal friendships leave behind him an influence which will be long felt by all who knew him and by many who never saw him.—[Memorial prepared by Joseph W. Roe, New York, N.Y. Mem. A.S.M.E.]

WALTER JOSIAH RICKEY (1871–1935)

Walter Josiah Rickey, managing director of The Singer Manufacturing Company, Ltd., Clydebank, Scotland, was seriously injured in a motor accident at Mantes, France, and died on May 22, 1935, ten days later, at Neuilly-sur-Seine, where he had been removed for specialized care.

Mr. Rickey had been connected with the Singer organization since October, 1903, when he became superintendent of the cabinet factory at South Bend, Ind. He remained there for about ten years, serving as works manager during the latter part of the time. He was made managing director at Clydebank in 1913.

The following excerpts are from the *Glasgow Herald* of May 24, 1935:

"His technical ability was outstanding, and his genius for organizing workshops and factories on mass-production lines greatly enhanced the reputation of the Singer Company, and made the Clydebank works among the most important of their kind in the world.

"His activities were not confined to the technicalities of production and administrative work, as he entered into welfare work with great enthusiasm. The welfare of the staff and workers was one of his first objectives when he came to Clydebank, and in cooperation with his codirectors he made provision for indoor and outdoor sport for the workers on a scale which cannot perhaps be rivaled by any large industrial establishment in this country.

"Mr. Rickey's enthusiasm for sports was of great help to him in organizing welfare schemes. He was a keen golfer and angler, and he was also interested in yachting."

He was particularly interested in the group insurance plan for factory workers and helped to put it into effect at the Clydebank works.

Mr. Rickey was born at Athol Center, Mass., on January 7, 1871, son of George Warren and Jane A. (Flint) Rickey. Graduated from the Massachusetts Institute of Technology in 1895 with an S.B. degree in mechanical engineering, he entered the employ of the T. & B. Tool Co., Danbury, Conn., as draftsman. Early the following year he was promoted to the position of chief draftsman and assistant superintendent. He left the company in January, 1898, to work in the manufacturing department of the General Electric Company at Schenectady, N.Y., and a year later was put in charge of the department. He continued in that position until he joined the Singer organization in 1903. He held a United States patent for a needle-straightening machine and methods.

Mr. Rickey took great interest in philanthropic and welfare work of all kinds. He was an active worker in the Boy Scout and Boys' Brigade movements. He was director of the Victoria Infirmary at Helensburgh from 1922 to 1925 and chairman for part of the time, and a member of the District Nursing Association there. He was a member of the County Committee at Clydebank for the British Red Cross, Scout Commissioner for Dumbartonshire, and a deacon in the Congregational Church at Helensburgh. During the last five years of his life he was a director of the Glasgow Chamber of Commerce.

Mr. Rickey became a member of the A.S.M.E. in 1903 and belonged to the Institution of Engineers and Shipbuilders in Scotland, and of the Glasgow and West of Scotland Association of Foreman Engineers. He was honorary secretary of the Massachusetts Institute of Technology in Glasgow. His clubs included the Anglo-American in London and a number of golf, yacht, and automobile clubs in Scotland. He was married in 1901 to Miss Grace Landon of Schenectady and is survived by her and by five daughters and a son.

WILLIAM F. RIPPE (1900–1935)

William F. Rippe, planning and installation engineer, Electric Generation Department of the Public Service Electric & Gas Co., Newark, N.J., died on August 24, 1935, at his home in West Orange, N.J., after a four months' illness. Mr. Rippe had been in the employ of Public Service since 1922, the year in which he was graduated from Cornell University, where he received his degree in mechanical engineering. He started with Public Service as a cadet engineer and, following the completion of that course in December, 1924, he was assigned to Marion Station as test engineer. He was transferred to Kearny Station as test engineer in September, 1926. On May 1, 1931, he was transferred to the general office as engineer and a month later was appointed fuel engineer. He was appointed chief engineer at Kearny Station, in June, 1934, and held that post until his promotion to planning and installation engineer, on May 15, 1935.

Mr. Rippe was born in New York, N.Y., on October 19, 1900. He prepared for college at the Mount Vernon High School. As an undergraduate at Cornell he was one of the best-liked students on the campus. His activities included captaincy of the varsity basketball team in 1922, and memberships in Tau Beta Pi, Pi Kappa Alpha, Phi Kappa Phi, Atmos, Sphinx Head, the College Honor Committee, and the Student Council. During his summer vacations he engaged in automobile-repair work, drafting, installation of machinery, production, and repairing for metal works, and bridge construction.

Mr. Rippe is survived by his wife, Gertrude, two children, Richard and Jane, his father, a sister, and a brother.

Mr. Rippe became a junior member of the A.S.M.E. in 1923.

CHARLES WILLIAM RIPSCH (1884–1936)

Charles William Ripsch, secretary and works manager of the Buckeye Portable Tool Company, Dayton, Ohio, died on February 1, 1936. He was born in that city on September 1, 1884, son of Cynthia Ann and Frank Ripsch. He attended the public schools of Dayton, including the Steele High School and Dayton Manual Training School and was graduated from Ohio State University with a degree in mechanical engineering in 1907. During his vacations he did drafting and designing on water wheels for the Stillwell-Bierce & Smith-Vaile Co. and the Corliss Engine and Pump Department of the Platt Iron Works Company, and during the latter part of his senior year carried on an investigation on the flow of gases through a venturi meter, making some six thousand observations on the flow of steam under varying pressures and working out formulas for steam and gas.

Following his graduation from the University Mr. Ripsch took a position as designer in the inventions department of the National Cash Register Company, Dayton, working on cash-register parts. Later he was made chief draftsman of the department, where he continued to be employed until the fall of 1910. After a trip abroad, on which he visited England, France, Belgium, Switzerland, and Germany, he became automobile designer for the Stoddard Dayton Automobile Company, but left there in May, 1911, to serve as inventor and tool designer for the Recording & Computing Machines Co., Dayton. Two years later he opened an office in Dayton for the practice of consulting engineering on inventions, special machinery design, and special tools for interchangeable manufacturing. At the expiration of another two years he became connected with the Dayton Pipe Coupling Company in the capacity of superintendent. In April, 1916, he was appointed secretary of the Sheffield Machine & Tool Co., Dayton, manufacturers of special machinery, dies, jigs, and tools, and subsequently he was chief engineer for the American Railways Equipment Company, and again connected with the Recording & Computing Machines Co., in the capacity of assistant chief engineer. For several years following the World War he was superintendent and chief engineer of the Joyce Cridland Company, Dayton. Since 1922 he had held his position as secretary and works manager of the Buckeye Portable Tool Company.

Mr. Ripsch had taken out a considerable number of patents in his own name and in the name of the companies for which he worked. These were principally on cash registers and their parts, air compressors and pumps, hoisting jacks, portable pneumatic and electric tools.

Mr. Ripsch became a junior member of the A.S.M.E. in 1908 and a member in 1916. He served as chairman of the Dayton Section of the Society in 1930 and was a member of the Dayton Engineers Club. Surviving him are his widow, Agnes L. (Batdorf) Ripsch, whom he married in 1929, and a daughter, Charlon B. Ripsch.

CHARLES EVERETT ROSS (1867–1935)

Charles Everett Ross, well known as a naval architect and marine engineer, died in New York, N.Y., on January 11, 1935. He was born in the Greenpoint section of Brooklyn, N.Y., on January 17, 1867, the son of Josiah Caldwell and Julia Bryant (Haines) Ross, who moved to Brooklyn from Maine about 1861. His mother's family were descendants of John Howland of the Mayflower group. Her grandfather owned pine forests in Maine and built several clipper sailing ships. The family lived at Damariscotta Mills, Maine. His father's family were early settlers of Ipswich, Mass., and moved to Nobleboro, Maine, soon after the Revolutionary War. His father's mother was a Caldwell of Ipswich. His father learned the ship joiners trade in Maine.

When the Roach shipbuilding yard was opened at Chester, Pa., Josiah Ross, with a number of men from a New York shipyard, moved there, and Charles received his preliminary education in the public schools of that city. He was graduated from the University of Pennsylvania with the degree of bachelor of science in 1888 and then took a one-year postgraduate course, receiving the degree of mechanical engineer in 1889.

As a college student, Mr. Ross worked in the shipyard during his summer vacations, and after graduation went to work there as a draftsman. He remained until 1899, the latter part of the time being chief draftsman in charge of the engine department. He then went to New York and until 1904 he was the engineer in charge of design and repair at the Morgan Iron Works there. Since then he had been in business in New York as a consulting engineer in naval architecture and engineering. For two years he was a member of the firm of Ross & Rodman and from 1906 until 1920, when he established his own office, he was associated with Frank S. Martin.

Mr. Ross became a member of the A.S.M.E. in 1900 and also belonged to The Society of Naval Architects and Marine Engineers, the Institution of Naval Architects (London), and the Downtown Athletic Club, New York. He was a New York State licensed engineer. He was musically inclined and was a student of Dickens and Shakespeare.

His wife, Katherine E. Dreemmy, whom he married in 1905, died in 1920. He had made his home at the Hotel Chelsea, New York, for many years. He is survived by a niece, Julia C. Ross, and a nephew, Elliott P. Ross. [Based, in part, on an obituary prepared for publication in the Transactions of The Society of Naval Architects and Marine Engineers.]

CARL ROSSMASSLER (1877–1935)

Carl Rossmassler, professor of machine design at Cooper Union, New York, N.Y., died suddenly from a heart attack on December 28, 1935, at his home in that city.

Professor Rossmassler was born in Philadelphia, Pa., on June 2, 1877, the son of Richard and Bertha (Collins) Rossmassler. He received his education at the Penn Charter School, Germantown, Pa., the St. Paul's Cathedral School, Long Island, N.Y., and the Massachusetts Institute of Technology, where he was awarded the B.S. degree in naval architecture in 1901.

After graduation from college, he was employed at the New York Shipbuilding Company, Camden, N.J., in hull work, and later at the Electro-Dynamic Company, Bayonne, N.J. In 1907 he was instrumental in forming the Rossmassler-Bonine Electric Company, in Philadelphia, manufacturing small electric motors and specialties, of which company he was treasurer. This organization terminated in 1913, after which he started the Morristown Boat and Engine Works, at Morristown, N.Y., where he engaged in the building of marine internal-combustion engines until 1916. War disturbances interfering with this business, he went with the St. Lawrence Marine Railway Company and the George Hall Coal & Transportation Co., Ogdensburg, N.Y., as mechanical engineer. From 1917 he spent two years in the Bureau of Construction and Repair, Navy Department, Washington, as supervising draftsman in charge of Standard Plans Division, and then short periods successively at the Durkee Manufacturing Co., Staten Island, N.Y., and the Domestic Engine & Pump Co., Shippensburg, Pa.

In 1920 Professor Rossmassler was appointed assistant professor of machine design at Cooper Union, New York, and in 1924 was made professor and placed in charge of the Department, which he had developed and made highly efficient. He held this position at the time of his death.

Professor Rossmassler became a member of the A.S.M.E. in 1921, and also belonged to the Society of Automotive Engineers and the Society for the Promotion of Engineering Education. He is survived by his widow, Lulie (Powers) Rossmassler, whom he married in 1922.—[Memorial prepared by E. F. Church, Jr., Brooklyn, N.Y. Mem. A.S.M.E.]

DANIEL ROYSE (1871–1935)

Daniel Royse, youngest son of Daniel and Martha D. (Coulson) Royse, was born on January 21, 1871, at Chauncey (now West Lafayette), Ind.

He attended the Lafayette public schools and in 1884 entered the preparatory department of Purdue University, from which he was graduated in 1889 with the degree of bachelor of mechanical engineering. The next two years he attended Cornell University (in 1890–1891 holding the Sibley fellowship) and in 1891 received the degree of M.M.E. The following year he was an instructor at Cornell and the next year was abroad in Germany and Switzerland. After his return he engaged in drafting and machine design at Purdue University, during 1894.

In 1895 Mr. Royse removed to Chicago and entered the office of David L. Barnes, consulting engineer. In December, 1896, he joined the staff of the Windsor & Kenfield Publishing Company as mechanical editor of the *Street Railway Review;* in 1901 he succeeded the late H. H. Windsor as editor in chief of the publications of that company, which also included *Brick and Steam Engineering.* In 1906 Mr. Royse sold his interest in this company and the following year became assistant editor in chief of *The Railway Age*, Chicago, in which position he continued until that journal was consolidated with the *Railroad Gazette.*

From 1906 to 1914 Mr. Royse was with the purchasing department of the Harriman Lines as assistant to the director of purchases of the Union Pacific Railroad and the Southern Pacific System, and, after the segregation of the Harriman Lines, as assistant to the vice-president of the Union Pacific Railroad, located in New York.

In 1915 he went to Indianapolis and for some three years was associated with his brother-in-law, Daniel B. Luten, in the National Bridge Company, returning to New York in 1918.

When in Chicago Mr. Royse engaged in the study of law and was graduated from the Chicago Law School in 1899. He was admitted to the Tippecanoe County bar in November, 1901, and to the Chicago bar in June, 1902.

Since 1923 Mr. Royse had been associated with Frank H. Hall, general counsel for the Corn Products Refining Company, New York. His death occurred at his home in that city on October 28, 1935.

Mr. Royse became a junior member of the A.S.M.E. in 1891 and a member in 1904. At Purdue he was elected to membership in the Kappa Sigma fraternity and at Cornell, to Sigma Xi, honorary scientific society. He belonged to the Engineers' Club, New York, and Union League Club, Chicago. He resigned as a major in the Ordnance Section, Officers' Reserve Corps, U.S.A., in 1926, and was a member of the Military Order of the Loyal Legion.

In 1910, Mr. Royse married Grace Heath (Hull) Marsteller, by whom he is survived.

JOHN MORRELL RUSBY (1861–1935)

John Morrell Rusby was born in Franklin, N.J., on March 5, 1861, son of John and Abigail E. (Holmes) Rusby. Following his graduation from Stevens Institute of Technology in 1885 he entered the Construction Department of the United Gas Improvement Company of Philadelphia, Pa., as cadet engineer. In April, 1886, he went to Allentown, Pa., to serve as superintendent of the gas works there and about a year later transferred to Jersey City, N.J., where he held a similar position. Later he took charge of the Hudson County Electric Plant at Jersey City and about 1899 was appointed engineer for the Hudson County Gas Company. He remained in Jersey City until July, 1902, when he returned to the main office in Philadelphia, where he served for a time as inspecting engineer and then engineer of tests. In this latter capacity he had charge of all the development work done by the company in the Chemical and Physical Laboratories, the Experimental Plant, and the Appliance Laboratory.

In connection with the research work carried on by the company under Mr. Rusby's direction he, individually and jointly with other company employees, was responsible for many important patents. Some of these were on processes of delayed combustion, blue-gas apparatus, steam and air controls, producer-gas operation, water-gas operation, vertical retorts, and waste-heat boilers. He retired from active professional work about six years prior to his death, which occurred at his home in Haverford, Pa., on May 27, 1935.

Mr. Rusby had been a member of the A.S.M.E. since 1893 and also belonged to the American Gas Association, the Illuminating Engineering Society, and The Franklin Institute, which awarded him its Longstreth Medal in 1914 for his paper on "Industrial Combustible Gases." He was a member of the Engineers Club, Philadelphia, and the Merion Cricket Club, Haverford. He is survived by his widow, Ella M. (Winans) Rusby, whom he married in 1910, and by a daughter, Jeanne Winans Rusby.

HARRIS JOSEPH RYAN (1866–1934)

Harris Joseph Ryan, professor emeritus of electrical engineering of Stanford University and honorary director of the Harris J. Ryan high voltage laboratory there, died on July 3, 1934. Professor Ryan was born at Powells Valley, Pa., on January 8, 1866, the son of Charles William and Louisa Mary (Collier) Ryan. He secured his early education at Baltimore City College and at Lebanon Valley College, Annville, Pa., and in 1883 entered Cornell University as a member of the first class in the newly organized course in electrical engineering.

During the year following his graduation in 1887, he worked on electric-light installations for the Western Engineering Company, Lincoln, Neb., then returned to Cornell to become an instructor in the department of physics. He was appointed assistant professor of electrical engineering in 1889, associate professor in 1892, and professor and head of the department in 1895 at the age of twenty-nine. He remained at Cornell until 1905, when he went to Palo Alto to become professor of electrical engineering, in charge of the department, at Leland Stanford Junior University, now known as Stanford University.

Shortly after becoming a member of the faculty at Cornell, Professor Ryan presented a paper before the American Institute of Electrical Engineers reporting work on transformers carried on by him and Ernest Merritt. This paper, which first brought him into prominence, was followed by others giving the results of his investigations in the electrical field. He developed the pole face winding and the principle of the interpole which counterbalanced the effects of armature reaction in direct-current machines. He also became interested in high-voltage transmission, and succeeded in disproving the then common belief that 40,000 volts was the maximum economical transmission voltage. In the course of this work he constructed an air-insulated transformer for 90,000 volts which was in use in the laboratory at Cornell for many years afterward. About 1900 he began a series of studies of the cathode-ray tube, which he applied to the measurement of voltage and current.

After going to California, Professor Ryan continued his researches in power transmission, and in recognition of his work the university built a high-voltage laboratory in 1913 in which the principal item of equipment was a 350,000-volt transformer. Here Professor Ryan constructed a 60,000-cycle oscillator for use in the study of insulator flashover, and in 1916 devised a potentiometer for measuring the potential across each unit in a suspension string.

In addition to his other work Professor Ryan was consulting engineer for the Los Angeles Aqueduct Power Bureau from 1909 to 1923 and during the year 1918–1919 he was director of the supersonics laboratory of the National Research Council in Pasadena. He devoted much time to the study of corona formation on conductors and was frequently called on as a consultant, particularly in connection with increasingly higher voltages.

The Harris J. Ryan high-voltage laboratory was opened in 1926. The building and equipment were given by manufacturing and power companies and the university provided the land, which included a strip suitable for a seven-mile experimental transmission line. The equipment included six 350,000-volt, 60-cycle transformers insulated to supply, with various connections, voltages up to 2,100,000 between lines, single-phase, 1,050,000 from line to ground, single-phase, and 1,212,000 between lines, three-phase. Professor Ryan retired from active duties at the laboratory and university in 1931.

Among the honors which came to Professor Ryan were the degree of doctor of laws by the University of California and the Edison Medal of the American Institute of Electrical Engineers, both in 1925. He joined the Institute as an associate in 1887, and was transferred to the grade of member in 1895 and to the grade of Fellow in 1923. He had been very active in committee work of the Institute, and served successively as manager, vice-president, and (1923–1924) president. In addition to the many papers presented before the Institute and before other societies, or published by them, Professor Ryan was coauthor, with H. H. Norris and G. L. Hoxie, of "A Textbook of Electrical Machinery," issued in 1906.

Professor Ryan had also served on the United States National Committee of the International Electrotechnical Commission and was United States delegate to the International Electrical Congress at St. Louis in 1904. He was a member of the jury on awards at both the World's Fair at Chicago in 1893 and the Panama-Pacific International Exposition at San Francisco in 1915.

Professor Ryan became a member of the A.S.M.E. in 1896, resigned in 1912, and was reinstated in 1923. He also belonged to the American Physical Society, the National Academy of Sciences, and the Institute of Radio Engineers and was a Fellow of the American Association for the Advancement of Science. He was a member of the honorary fraternities, Sigma Xi and Tau Beta Pi, as well as of Phi Kappa Psi.

He is survived by his widow, Katherine E. (Fortenbaugh) Ryan, whom he married in 1888.

CHARLES ELLIOTTE SARGENT (1862–1934)

Charles Elliotte Sargent, who was awarded the John Scott Medal for his complete-expansion gas engine and who had taken out some seventy patents covering a variety of inventions, died at his home in Indianapolis, Ind., on September 22, 1934, of a cerebral hemorrhage.

Mr. Sargent was born at Carlinville, Ill., on September 12, 1862, the son of Jacob True and Maria L. (Braley) Sargent. He attended the local district schools and Blackburn College, from which he was graduated with a B.S. degree in 1882. He received a B.S. degree in mechanical engineering from the University of Illinois in 1886, and also an M.S. degree from Blackburn. The University of Illinois conferred an M.E. degree upon him in 1915.

During summer vacations from 1883 to 1886 and for two years following his graduation from the University of Illinois he worked for the Walter A. Wood Co., of St. Louis, Mo., manufacturers of mowing and reaping machines. He then entered the employ of A. L. Ide & Sons, Springfield, Ill., spending the first year with the company in the erection of steam plants. He next became chief draftsman for the company and two years later was made manager of its Chicago office. He tested steam-plant equipment for economy and efficiency, and served as consulting engineer on the design and erection of steam plants. He also engaged in the sale of simple and compound steam engines manufactured by the company.

From 1894 to 1898 Mr. Sargent was manager of the Ball & Wood Co., Chicago, and during the next five years was located at the Elmes Engineering Works in that city. It was here that he built his first complete expansion double-acting gas engine (which was described in a paper presented before the A.S.M.E. in December, 1900, and published in the Transactions for 1901). The Sargent Gas Engine Company, of Chicago, was established in connection with this invention.

During a period of consulting work at this time, Mr. Sargent served the Wellman-Seaver-Morgan Company of Cleveland, Ohio, the Ball & Wood Co., and the Wisconsin Engine Company, of Corliss, Wis. He was chief engineer for this company for the years 1908–1910.

In 1914 Mr. Sargent went to Indianapolis, Ind., to take the position of chief engineer of the Lyons Atlas Company, which about five years later became the Midwest Engine Company. He continued with his company until 1920, and from then until he retired in 1931 engaged in research engineering for various companies.

Throughout his life Mr. Sargent was engaged in research and invention under his own name. The list of patents held by him includes many entries for internal-combustion engines. The problem of a constant compression complete-expansion internal-combustion engine engaged his attention at an early date, as did also valve mechanisms, and many of his patents are in this field. His automatic gas calorimeter, as well as his steam meters, have won special recognition. Other patents show an interesting diversity of research. They include windmill generators and governors, a farm-tractor control device, boiler-feedwater purifiers, a dust separator, an electric cooker, card-cutting machinery, pressure and draft gages, and a gas washer.

Mr. Sargent's contributions to the technical press, including the publications of the societies to which he belonged, were numerous.

Mr. Sargent was given the John Scott Medal in 1907. In 1905 he was presented the Octave Chanute Medal of the Western Society of Engineers for his research work in thermodynamics. In addition to his membership in that Society he belonged to The Franklin Institute and the Society of Automotive Engineers. He had been a member of the A.S.M.E. since 1891. He held the 32d Masonic degree and was a Shriner and a member of the Scottish Rite.

Surviving Mr. Sargent are his second wife, Harriette M. (Boynton) Sargent, whom he married in 1895, and three children, Richard B. Sargent, Mrs. Francelia (Calvin R.) Hamilton, and Mrs. Charline (Ross C.) Lyons. His first wife, Alice (Vaughan) Sargent, died in 1894, four years after their marriage.

CARL HUGO SCHUTTLER (1887–1935)

Carl Hugo Schuttler was born at Wheeling, W.Va., on May 22, 1887, son of Joseph and Caroline Louisa (Weiderbusch) Schuttler. He attended the grade and high schools of Wheeling and then served a two-year apprenticeship as a machinist with Spears & Riddle, in Wheeling, at the same time completing the mechanical-engineering course of the International Correspondence Schools.

From 1906 to 1909 he was employed by the Whitaker Glessner Company in its Corrugating Department at Wheeling. During the early part of the time he was a detailer in the Engineering Department, but later, during a seven-month leave of absence of the me-

chanical engineer, he was given the responsibility for all maintenance work in the mill and for the completion of extensive additions to their plant at Martins Ferry, Ohio.

In 1909 Mr. Schuttler was engaged by the Wheeling Mold & Foundry Co. to design rolling-mill machinery. Later he was put in charge of the construction and equipment of their steel foundry, and subsequently developed shop-production methods for the execution of contracts for the Isthmian Canal Commission.

Thus it was with a considerable amount of experience that Mr. Schuttler, at the age of twenty-five, entered Lehigh University to secure a more thorough technical training. He was graduated with an M.E. degree in 1916.

In March, 1918, Mr. Schuttler became assistant district supply manager, at Youngstown, Ohio, for the United States Shipping Board, Emergency Fleet Corporation. He was in full charge of the production of marine turbines and engines and deck machinery. During the latter part of the year he attended the Field Artillery Central Officers Training School, at Camp Zachary Taylor, Kentucky, returning to Youngstown in January, 1919, to take the post of senior examiner on cancelations. He had full charge of surveys of the cost of manufacture on canceled contracts for marine engines and turbines and deck machinery.

Upon the completion of this work Mr. Schuttler took up private practice at Toledo, Ohio, as industrial engineer and public accountant, in which he engaged until 1928, part of the time as senior member of the firm of Baker & Schuttler. He then spent a year as engineer for the Mineral Felt Company, Toledo, and several years as engineer and sales manager for the Insulating Products Company, Aurora, Ill. His work there led to a patent on heat-insulating material.

In 1934 he returned to Wheeling to become general foreman of the Drop Forging Department of the Wheeling Corrugating Company, a subsidiary of the Wheeling Steel Corporation, the position held by him at the time of his death on July 28, 1935.

Mr. Schuttler is survived by his widow, Estelle (Vinson) Schuttler, whom he married in 1926, and by three children, Walter Schuttler, Dorothy (Schuttler) Brown, and Dorothea (Schuttler) Cromer, all of Aurora, Ill.

As a student at Lehigh University, Mr. Schuttler was elected to membership in the Kappa Sigma fraternity. He became an associate-member of the A.S.M.E. in 1918 and a member three years later. He was also a former member of the Engineers' Society of Western Pennsylvania.

GEORGE HASKELL SCOTT (1851–1935)

George Haskell Scott, a member of the A.S.M.E. since 1893, died on October 29, 1935, after a brief illness of pneumonia. He was past his eighty-fourth birthday, having been born on June 29, 1851, at Hartland, Conn. His parents were Nelson and Martha (Gaylord) Scott. The family moved to Edgartown, Mass., when he was a small boy, and later to Marblehead, Westboro, and Amherst. He completed his secondary education at Amherst and entered the Worcester Polytechnic Institute in 1869. After receiving his B.S. degree in mechanical engineering in 1872 he remained at the Institute for several years as assistant in the woodworking department.

For short periods after leaving the Institute Mr. Scott was employed as a patternmaker in West Fitchburg and as a machinist in Worcester. He then took a position with the Washburn & Moen Manufacturing Co., Worcester, as draftsman and machinist, working on the improvement of special machines. He was with this company until 1881, when he became superintendent of the Morgan Spring Company, Worcester, a position which he held until 1890. He then returned to the Washburn & Moen Manufacturing Co. as master mechanic and was connected with the plant until the early part of 1900, after it was acquired by the American Steel & Wire Co.

During the next five or six years he was again associated with the Morgan Spring Company as superintendent and while there invented a number of machines for making springs. In 1908 he was treasurer of the Worcester Injector & Valve Co., and the following year was vice-president of the Dodge Sander Company, Worcester.

Mr. Scott became an instructor at the newly organized Worcester Trade School in 1910 and after a few years there took the position of superintendent of grounds and buildings at the Worcester Academy. In 1914 he was manager of the Union Machine Company, Worcester, and in 1917 went to Bridgewater, Mass., where he assisted a friend in the establishment of the Chandler Construction Company. During the next few years he worked for the Carver Cotton Gin Company, East Bridgewater.

In 1922 Mr. Scott went to Merchantville, N.J., to live with his son, Edgar G. Scott, later moving with him to Moorestown, N.J. Although not regularly employed he had kept active in a small repair shop of his own until shortly before his death.

Mr. Scott's wife, formerly Miss Ella Marie Clark, of Worcester, died in 1913. Only the one son, who is also a member of the A.S.M.E., survives him.

DAWSON HANCOCK SKEEN (1887–1936)

Dawson Hancock Skeen, president of D. H. Skeen & Co., Chicago, Ill., power equipment sales engineers, died at his home in Flossmoor, Ill., on February 23, 1936, after a week's illness, of pneumonia.

Mr. Skeen was born at Bell Buckle, Tenn., on October 6, 1887, the son of Matthew L. and Ida I. Skeen. He attended the Webb Preparatory School at Bell Buckle and was graduated from the United States Naval Academy at Annapolis in the Class of 1910. He resigned from the Navy upon the death of his father a few months later and after closing out his father's business took a position with the National Light & Power Co., at Fulton, Ky.

When the United States entered the World War in 1917 Mr. Skeen reentered the service as senior lieutenant in the Bureau of Steam Engineering, Navy Department, at Washington. After the War, in 1919, he went to Chicago to become assistant to the president of The Edward Valve & Manufacturing Co., later becoming superintendent of the works of that company at East Chicago, Ind. He organized D. H. Skeen & Co. in 1922 and from then until his death was president of the company, representing leading manufacturers of power-plant equipment.

Mr. Skeen became a member of the A.S.M.E. in 1927 and served as chairman of the Chicago Local Section from July, 1931, to July, 1933. He also belonged to the Western Society of Engineers and was a director of the Flossmoor Country Club. He is survived by his widow, Elizabeth C. Skeen, whom he married in 1915, and by two children, Nancy Lee and Drake H. Skeen.

JOHN SKOGMARK (1877–1935)

John Skogmark, chief engineer for Chemical Appliances, Inc., New York, N.Y., died at the Swedish Hospital in Brooklyn, N.Y., on March 26, 1935. He was a native of Sweden, having been born at Ore on May 6, 1877, son of Andera and Karin (Dunder) Skogmark, but was a naturalized citizen of the United States.

Following his graduation as a mechanical engineer from the Chalmers Institute of Technology in 1897, Mr. Skogmark spent about a year with the Jonsered Manufacturing Company, at Jonsered, Sweden. From then until the end of 1902 he was engaged in waterworks construction, for the cities of Gothenburg and Falkenberg.

His first position in this country was as machine designer at the Industrial Works in Bay City, Mich. For a year beginning in July, 1904, he held the position of master mechanic at Culebra for the Isthmian Canal Commission. After returning to the States he was employed for about six months as machine designer for The Bucyrus Company, at South Milwaukee, Wis., and for part of 1906 in powerhouse construction at the Raritan Copper Works, Perth Amboy, N.J. He then went to Peru, S.A., to work for the Cerro de Pasco Mining Company, at La Fundicion, as assistant engineer of smelter construction.

He returned to Wisconsin in 1909 and for a number of years was construction engineer for the Mineral Point Zinc Company. Then he went to New York, where he engaged in consulting practice until he became affiliated with Chemical Appliances, Inc. He helped to design a munitions plant at Nitro, W.Va., during the World War period. He held patents on a precipitating apparatus and process, a process and apparatus for extracting zinc ore, and a process for the purification of gas.

Mr. Skogmark had been a member of the A.S.M.E. since 1912. He also belonged to the American Institute of Mining and Metallurgical Engineers, was a past-president of the American Society of Swedish Engineers, and was vice-president of the John Ericsson Society at the time of his death. He was a member of the American Scandinavian Foundation.

Mr. Skogmark was unmarried.

SYLVESTER MILLER SNOW (1861–1935)

Sylvester Miller Snow, who died in Providence, R.I., on April 28, 1935, was born in that city on September 3, 1861, the son of Edwin Miller Snow, M.D., and Ann Eliza Warren (Pike) Snow. He prepared for college at the University Grammar School, Providence, and entered Brown University in 1879. After receiving an A.B. degree in 1883 he became a draftsman with the Providence Steam Engine Company. In 1886 he received an A.M. degree from Brown, and in the same year went to the Brown & Sharpe Manufacturing Co. as a draftsman. He left there in 1888 to enter the employ of the Corliss Steam Engine Company, and in 1889 became head draftsman of the

Wm. A. Harris Steam Engine Co. From 1890 to 1893 he was mechanical engineer for the Manufacturing Investment Company, Madison, Me. He returned to Providence in 1894, engaging in business there until 1898, when he took up mechanical engineering work in Montreal, Que.

In 1899 Mr. Snow became senior partner in the firm of Snow and Humphreys, mechanical and mill engineers, with offices in Boston, a partnership which continued until 1910. Since that time Mr. Snow had been in business for himself as a mechanical engineer in Boston and Providence.

From 1885 to 1887 he served as clerk of the Warren Baptist Association and wrote the annual reports published in 1885–1887. He had been a member of the A.S.M.E. since 1890. A bachelor, he is survived by a sister, Elizabeth Hathaway Snow.

GRANT WARREN SPEAR (1865–1935)

Grant Warren Spear, vice-president of the Dearborn Chemical Company, died at Palm Beach, Fla., on March 22, 1935, after several months' illness. He is survived by his widow, Mary (Carr) Spear, whom he married in 1890, and by a daughter, Emily Lenore Spear.

Mr. Spear was born at Aurora, Ill., on April 7, 1865, son of Warren and Matilda (Griffith) Spear. He was graduated from the West Aurora High School in 1883 and from the University of Illinois with a B.S. degree in mechanical engineering in 1887. He became a member of the firm of Warren Spear & Co. following his graduation and after the death of his father was president of the company until he joined the Dearborn Chemical Company, of Chicago, in 1898. He was testing engineer of this company for several years, later sales manager, then vice-president. In 1908 he went to New York to serve as eastern manager. He had charge of all business in the East and all foreign and export business, supervising both sales and engineering.

Mr. Spear became a member of the A.S.M.E. in 1914. He belonged to the Northport Country Club and Crescent Athletic Club, Huntington, Long Island, the New York Railroad Club, the Chicago Athletic Club, and the Havana Country Club.

FREDERICK LEWIS STANTON (1892–1934)

Frederick Lewis Stanton, whose death occurred on January 28, 1934, was born in Guysboro County, Nova Scotia, on January 25, 1892, son of Richard and Emma (McKenzie) Stanton. He served a full apprenticeship from 1909 to 1912 with the W. J. Young Machinery Co., Lynn, Mass., and after working for a few months on designing for the Peerless Automobile Company, Boston, spent nearly a year in the employ of the New England Gear Works in that city as a machinist, securing a practical knowledge of gearing.

From 1913 to 1915 Mr. Stanton was connected with the Associated Tanners Machinery Company, Salem, Mass., designing machinery, and, during the latter part of the time, being in charge of the drafting room. Subsequently he spent about two years in machine design and experimental work for the Essex Machine Company, Lynn, and a year with the United Shoe Machinery Company, Beverley, Mass., in the Tool Designing Department, designing jigs and fixtures. He became a naturalized citizen of the United States in 1917.

Since 1918 Mr. Stanton had been with the National Paper Products Company, Carthage, N.Y., in charge of machine designing and the experimental shop.

Mr. Stanton supplemented his early public school education in a number of ways. He studied mechanical drafting in the evening schools at Lynn, Mass., for four years and also took a two-year preparatory course there with the intention of entering the Massachusetts Institute of Technology. He gave up this idea when the opportunity at Carthage was offered him, but continued his studies in mechanical drafting, mathematics, and mechanical engineering through the International Correspondence Schools.

Mr. Stanton became an associate-member of the A.S.M.E. in 1928, and was a Mason. He is survived by his widow, June (Spearance) Stanton, and by two children, Dorothy E. and Richard L. Stanton.

CHARLES WOOD STEPHEN (1885–1935)

Charles Wood Stephen, chief engineer of the Reading-Pratt & Cady Co., Inc., Bridgeport, Conn., died of pneumonia at the Bridgeport Hospital on May 2, 1935.

Mr. Stephen became an associate-member of the A.S.M.E. in 1916 and a member three years later and had rendered the Society valuable service in Local Section work in Bridgeport and on technical committees. He was a member of the Executive Committee of the Bridgeport Section in 1929, and had been one of the Society's representatives on the Sectional Committee on the Standardization of Pipe Flanges and Fittings since its organization in 1921. He served on two subcommittees of the committee—those on Cast Iron Flanges and Flanged Fittings and on Screwed Fittings—and was chairman of a subgroup on the Design of Bolted Flange Connections.

He was also a member of the Manufacturers Standardization Society of the Valves and Fittings Industry and served on many of its committees, in several instances as chairman. A memorial resolution passed by the society referred to him as having been "associated with the valve and fittings industry for a quarter of a century and for the past decade or more a faithful and active worker in this society. . . . Through his engineering ability he has contributed substantially to the advancement of the art of our industry."

Mr. Stephen was born in Glasgow, Scotland, on December 23, 1885, son of John George Airlie and Edith (Lincoln) Stephen. He came to the United States in 1907 and was naturalized in 1916.

His technical education was secured at the Emmanuel School and South Western Polytechnic, both in London, between the years 1902 and 1906. He also served a three-year apprenticeship with Aiton & Co., Engineers, beginning in January, 1903. This included both shop and drafting experience, particularly in connection with pipe work, and during the latter part of his apprenticeship he was in charge of piping erection at the Battersea power house.

Mr. Stephen came to the United States by way of Canada, where he was employed during the summer of 1907 as a machinist by the Canadian Pacific Railroad, repairing locomotives at Montreal and Toronto. His first position in this country was with the Yetman Typewriter Transmitter Company at North Adams, Mass., working out some improvements in design. Subsequently he was employed by the Union Typewriter Company in Bridgeport to design tools for making the visible model then being perfected by the company. He next spent periods of about six months with the Locomobile Company of America and the Bullard Machine Company in design work, and a short time selling printing machinery for the Simplex Manufacturing Company, all in Bridgeport.

In May, 1910, Mr. Stephen became draftsman and designer for the Crane Company in Bridgeport. About three years later he was put in charge of the Experimental and Test Department, and during his last year with the company, 1916–1917, was in charge of the Specialty Department, manufacturing regulating valves and other steam specialties.

In July, 1917, Mr. Stephen went to Hartford, Conn., to take the position of mechanical engineer for Pratt & Cady, Inc. During his first year and a half there he was responsible for the design and inspection of all products manufactured by the company. He was appointed mechanical engineer and assistant works manager in January, 1919, and continued in that capacity until 1921, when he returned to Bridgeport to take the post of research engineer for the Reading-Pratt & Cady Co. Except for a year (1923–1924) with Jenkins Bros., Bridgeport, he continued with the company until his death, being made chief engineer in 1924.

Mr. Stephen patented a lubricant seal type of valve which has proved its worth in service. He was of a studious, retiring character, enjoyed golf and fishing, and was a member of the Brooklawn Country Club and of the Masonic fraternity. He is survived by his widow, Edith (McWilliams) Stephen, of Bridgeport, whom he married in 1912.

HERMANN STEPHENSON (1885–1934)

Hermann Stephenson, who died on March 27, 1934, was born in Los Angeles, Calif., on February 18, 1885, son of Walter Dorrence and Susan Cleveland (Utley) Stephenson. He prepared for college at the Ithaca High School and was graduated from Cornell University with an M.E. degree in 1907. He spent the summer of 1904 working in the shops of the American Locomotive Company at Schenectady, N.Y., and the following summer with the Wire & Telephone Co. of America, Rome, N.Y., as draftsman.

Following his graduation in 1907 Mr. Stephenson worked for a year on design, drafting, and field construction for Westinghouse, Church, Kerr & Co., New York, N.Y. During the next two years he was instructor in machine design and experimental engineering at Cornell University, and he returned there for the year 1915–1916 to establish a course in industrial management. In the interim he was engaged in efficiency work for the Brighton Mills, Passaic, N.J.

From October, 1916, until his retirement in 1926, Mr. Stephenson was employed by the Eastman Kodak Company at Kodak Park, Rochester, N.Y., the largest plant of the company, in the capacity of industrial engineer. His duties involved important engineering investigations in connection with manufacturing problems.

Mr. Stephenson became a junior member of the A.S.M.E. in 1909 and an associate-member ten years later. He belonged to the Knights Templar.

JAMES DANIEL STEWART, JR. (1904–1934)

James Daniel Stewart, Jr., son of James Daniel and Alula (Speight) Stewart, was born at Savannah, Ga., on July 12, 1904. He was graduated from the high school at Magnolia, Ark., in 1921, and attended the Georgia School of Technology, Atlanta, the following year. He then transferred to Clemson College, in South Carolina, from which he received a B.S. degree in mechanical engineering in 1925. He served as chairman of the A.S.M.E. Student Branch at Clemson during his senior year.

Offered a number of positions when he graduated, Mr. Stewart decided to go with the Atlantic Coast Line Railroad Company, for which his father was a machinist. He served a special apprenticeship at the Emerson Shops at Rocky Mount, N.C., and in June, 1927, was appointed plant engineer there. His engineering training and practical experience enabled him to solve many technical problems which arose both at the shops and elsewhere on the road. Changes in personnel led to his transfer in March, 1933, to the position of assistant power-plant engineer, though his duties continued to be much the same as before. Just prior to the attack of pneumonia from which he died on April 26, 1934, he had been assigned the task of air-conditioning the cars for the line. He had invented a portable journal jack and an oil-reclaiming machine. For six years he taught engineering classes at the Y.M.C.A. in Rocky Mount, many of his pupils being apprentices at the Emerson Shops.

Mr. Stewart became a junior member of the A.S.M.E. in 1929. He was elected to the Phi Gamma Delta fraternity at the Georgia School of Technology, and was a member of the First Baptist Church at Rocky Mount, and a second lieutenant in the infantry.

Surviving Mr. Stewart are his widow, the former Miss Sybil Ray, of Florence, S.C., whom he married in 1925, and two sons, James Ray and Charles Daniel Stewart.

FRANK H. STREINE (1878–1935)

Frank H. Streine, a member of the A.S.M.E. since 1919, was instantly killed in an automobile accident on June 14, 1935.

Mr. Streine was born in Leipsic, Germany, on November 17, 1878, the son of Frank Edward and Emily Emma Streine. He was a citizen of the United States, however, by virtue of his father's naturalization. His early education was secured in the public schools of Newport, Ky., and his technical training at the Ohio Mechanics Institute, Cincinnati, where he spent two years. He served an apprenticeship with the G. A. Gray Co., Cincinnati, planer manufacturers, and worked for the company as a journeyman until 1901. During the next six years he was employed by the Crawley Book Machinery Company at Newport as a journeyman machinist and from then until 1912 he worked in different shops in Cincinnati to increase his knowledge of other types of machinery. In 1912 and 1913 he was superintendent of construction of water works at Newport, having charge of the rebuilding of several large reservoirs and a pump house, and the installation of a water purification plant.

Returning to Cincinnati in 1914, Mr. Streine became chief draftsman for the Bickett Machine & Manufacturing Co. there, and subsequently he was made assistant manager of the plant and helped to bring out several machines for work on paper goods, as well as milling machines and planers. With all of this experience behind him, he organized the Streine Tool & Manufacturing Co., in New Bremen, in 1918, and served as its vice-president and general manager until 1934, when the Mill Equipment Company, of New Bremen, was organized with Mr. Streine as president and general manager. At the time of his death he was manager of the Sheet and Strip Finishing Machinery Division of the Mackintosh-Hemphill Company, of Pittsburgh, Pa. He held several patents on metal-shearing machinery.

Mr. Streine was keenly interested in education. He was founder of the New Bremen Parent-Teacher Association and served terms on both the local and county boards of education. He belonged to the Ohio Chamber of Commerce, the Civic Association and Square and Compass Club in New Bremen, and to the Masonic Order, Order of the Eastern Star, and Knights of Pythias.

Surviving Mr. Streine are his widow, Caroline Louise (Berndt) Streine, whom he married in 1918, and four children by a previous marriage, Dorothy (Mrs. Clyde) Griffin, and Frank, Elizabeth, and Carol Streine. His first wife, Emma Clara Berndt, died in 1917.

JOHN CHARLES STROTT (1894–1935)

John Charles Strott, assistant to the electrical engineer of the Consolidated Gas Electric Light & Power Co. of Baltimore, Md., died suddenly on June 16, 1935, at his home in that city. His widow, Rosetta (Whitney) Strott, survives him. Mr. Strott was born in Baltimore on October 15, 1894, and received his technical education at the Baltimore Polytechnic Institute and the Maryland Institute, from which he was graduated in 1913.

During his school period and the two years following his graduation he was employed successively in the Electric Department of the Consolidated Gas Electric Light & Power Co.; as draftsman for the York Engineering Company, York, Pa.; as designer and inspector for the G. E. Painter Co., Baltimore, working on power plants, heating and ventilating, electrical and plumbing equipment; similar work for the James Posey Company; and as assistant shop engineer for the J. G. Brill Co., Philadelphia, in connection with the manufacture of ammunition.

In October, 1915, Mr. Strott became assistant works engineer for the U.S. Industrial Alcohol Co., Curtis Bay, Md., with whom he had risen to the position of chief engineer in 1917, when he entered the U.S. Navy, Bureau of Yards and Docks, Washington, D.C. In the following year he served on the staff of the Bureau of Aircraft Production and was engineer inspector until early in 1919, when he became chief of construction of the Wrigley Chemical Plant of the Bon-Air Coal & Iron Corp. at Lyles, Tenn. In the fall of 1919 he returned to Maryland to serve as chief engineer of the American Cellulose & Chemical Manufacturing Co. at Cumberland. For a short period in 1922 he worked on the perfection of a hand stoker for the Patterson Hand Stoker Corporation, Baltimore. Since September, 1922, he had been connected continuously with the Engineering Department of the Consolidated Gas Electric Light & Power Co. of Baltimore as designer of power plants and assistant to the electrical engineer.

Mr. Strott became a junior member of the A.S.M.E. in 1916 and an associate-member in 1925. He served as chairman of the Baltimore Section during the year prior to his death and had been reelected to that office for another year. He was also a member of the American Institute of Electrical Engineers and the Baltimore Engineers Club.

An obituary published in *The Baltimore Engineer* for August, 1935, includes the following paragraph:

"Mr. Strott combined a broad knowledge of engineering with a kindly and cooperative attitude toward his fellow men. His fine civic spirit, and the confidence in and appreciation of him by his fellowmen, is evidenced by the leading part he took in the work of the Emergency Relief Committee of Technical Societies in Maryland and by his selection to the presidency of his neighborhood improvement association. He was an active worker for the Engineers Club and was highly esteemed. We shall mourn him as a true friend and ever helpful associate whose sterling qualities will long be remembered."

CARL SULZER-SCHMID (1865–1934)

Carl Sulzer-Schmid, president of Sulzer Brothers Limited of Winterthur, Switzerland, and Member of the National Council in the Swiss Parliament since 1917, died at his home at Winterthur, Switzerland, on October 30, 1934. Dr. Sulzer had been a member of the A.S.M.E. since 1891.

Jakob Carl Sulzer, the son of Heinrich and Bertha Sulzer-Steiner, was born in Winterthur, Canton of Zurich, on the 4th of February, 1865. His father, Heinrich, was the eldest son of Johann Jakob Sulzer, one of the founders of the famous engineering works bearing the Sulzer name now over one hundred years old. He attended the local schools, graduating from the high school at Winterthur in 1882 and the Académie in Lausanne in 1884. After his first military training he spent two years in the Sulzer Bros. shops acquiring skill as a molder, boilermaker, blacksmith, and machinist. Entering the Royal Polytechnicum at Dresden, Germany, in 1887, he graduated from the engineering course in 1889.

Returning to Sulzer Bros. he spent the next two years in Germany, Austria, England, and other European countries erecting and testing engines, boilers, and ice machines made by that company. In the fall of 1890, he traveled extensively in Canada and the United States, ending his American experience with five months at the Brown & Sharpe factory at Providence, R.I. It was during this period that his application for full membership was presented to the Society by H. A. Hill of Hill, Clark & Co.

Returning to Winterthur in June, 1891, he entered into active participation in the management of Sulzer Bros., becoming a partner in the firm in 1895. In the next 20 years he took a leading part in the development of the business, and on the conversion of the partnership into a limited liability corporation in 1914 was elected president of the company, which position he held at the time of his death.

At the time he became an active factor in the development of Sulzer Bros., boilers, engines, and ice machines represented the largest part of their product. The foundry was large and did much for other manufacturing concerns. He took a leading part in engineering developments of his firm, especially in boilers, steam engines, and centrifugal pumps, and many other lines of manufacture were soon added to the products of the firm. In the 40 years of his management the

plant grew from about 2000 hands to over 6000, while the productivity increased much more. He was particularly interested in the welfare of the workmen, and would point out three generations of one family at work in the shop. He told the writer that he knew personally every employee, their wives, and many of the older children. His personal interest was also shown in the schools, cooperatives, and local welfare societies.

He was an influential member of the Swiss Boiler Owners' Association for 30 years and since 1913 had been very active in the Swiss Association of the Employers in Engineering and Metal Industries. He had been president of the Swiss Association of Machinery Manufacturers since 1915. He was a member of the Zurich Chamber of Commerce and of the Swiss Chamber of Commerce. He became a member of the Verein deutscher Ingenieure in 1899, was a recipient of the V.D.I. medal, and later a director of the Deutsches Museum. In 1927 he received from the Federal Technical University in Zurich the honorary degree of doctor of engineering. Dr. Sulzer was most sympathetic toward international standardization and took an important part in the formation of standard test codes and other engineering standards.

Dr. Sulzer made many firm friends in the few months of his American visit. These friends he kept and the list grew, for the welcome at Winterthur was warm and he delighted in showing courtesies to his American visitors. His methods of thinking were as familiar to his American friends as his speech. He was equally at home in the three Swiss languages.

An ardent patriot, he gave freely of himself in the councils of the republic, the affairs of the trade organizations, chambers of commerce, and technical societies. Conservative in scientific and technical matters, in politics, and in trade, he was ever ready to recognize merit in newer and better ideas, methods, and materials, while his ripe judgment, wide experience, and rugged honesty kept him clear of fallacies and change without solid reason. His wide sympathy endeared him to his friends and neighbors, his associates in the manufacturing trade, and indeed all who knew him.

He is survived by his wife, two sons, and four daughters, and a number of grandchildren.—[Memorial prepared by Geo. A. Orrok, New York, N.Y., Mem. A.S.M.E.]

EVERETT WILES SWARTWOUT (1885–1935)

Everett Wiles Swartwout was born at New City, Rockland County, N.Y., on September 9, 1885, the son of Frank Green and Margurette E. (Wiles) Swartwout. He attended the Scarsdale School and White Plains High School and was graduated from Columbia University in 1908 with a degree in mechanical engineering. He was a member of Tau Beta Pi, honorary engineering society, and of the Delta Upsilon fraternity.

Mr. Swartwout's engineering experience began in the summer of 1907, during the early part of which he worked for the Union Railroad (Tarrytown, White Plains, and Mamaroneck), repairing and overhauling machinery at the power house in White Plains. Later in the summer he was on the maintenance force at the Hotel Astor, New York. During the summer following his graduation he worked on multivane exhaust and pressure blowers for the Sirocco Engineering Company, New York, and from October first until early in 1909 was purchasing agent for that company. The remainder of that year was spent in the employ of the American Blower Company, New York, as sales engineer, and the A. D. Granger Co., also of that city, in connection with the erection of engines, boilers, steam piping, stacks, and power plants in general; he was draftsman and assistant superintendent of construction during the first month and subsequently superintendent of construction.

In 1910 Mr. Swartwout began a period of ten years' employment with the Nordberg Mfg. Co. He engaged in sales work in the New York office until August, when he went to Milwaukee to spend the remainder of the year in the shops and engineering department of the company. He then took up sales work again and in July, 1911, was appointed manager of the Chicago office. In 1915 he returned to the New York office, assigned to sales and engineering in that territory. He was transferred to the Philadelphia office as manager in 1918 to handle business for the Emergency Fleet Corporation and other special war sales and engineering work. This office was closed on December 31, 1918, and Mr. Swartwout terminated his connection with the Nordberg Company on August 31, 1919.

During the next two years he was associated with the National Aniline & Chemical Co., part of the time as test superintendent at the Marcus Hook, Pa., plant, and the remainder as assistant construction engineer at the home office of the company in New York.

In 1921–1922, Mr. Swartwout assisted the Engineering Societies Employment Service of the four national engineering societies in organizing the Volunteer Committee, a group of members of the societies in New York devoted to canvassing employers for positions for unemployed engineers. Mr. Swartwout served as the first chairman of the group and subsequently as chairman of its executive committee. He also represented the A.S.M.E. on the "Joint Committee of Eight" of the Engineering Societies Employment Service.

In addition to the private research and development in the years following the close of the War, Mr. Swartwout did some work for the Lamson Company, Arca Regulators, Inc., and the K. B. Pulverizer Co. In 1924 he became manager of the New York office of the Connersville Blower Co. When this company merged into the Stacey Engineering Company he retained the position of New York manager, and when the Stacey company merged into the International Derrick & Equipment Co. of Columbus, Ohio, in 1931, he became chief engineer, division sales manager, and special negotiator for the new company, the International Stacey Corporation, with headquarters at Columbus. His position was terminated through the action of a creditors committee in March, 1932, and he was not regularly employed until September, 1934, when he entered upon inventory work for the Westchester Lighting Company. This continued until about a month before his death on December 30, 1935.

Mr. Swartwout made recognized notable improvements in signaling attachments for automobiles, uniflow steam engines, positive pressure blowers and gas pumps, flexible couplings, and meters transmuting, mechanically or electrically, variable pressure and/or temperature values into values under standard conditions.

Mr. Swartwout became a junior member of the A.S.M.E. in 1910, an associate member three years later, and a member in 1922. He was a representative of the Society at a conference on "Colors for Traffic Signals" called by the American Engineering Standards Committee in 1922. He was also a member of The Society of Naval Architects and Marine Engineers, American Gas Association, White Plains Congregational Church, the Columbia and White Plains University Clubs, and the Masonic fraternity.

Mr. Swartwout married Amy Ruden Conklin of White Plains, N.Y., in 1912, and is survived by her and by two children, Betsey and Everett W. Swartwout, Jr.

FRANK HENDRICKSON TAYLOR (1855–1934)

Frank Hendrickson Taylor was born at Cincinnati, Ohio, on November 20, 1855. His father, David Hendrickson Taylor, was descended from Matthew Taylor, who came to this country from England in 1664 and settled in East Jersey. His mother, Laura Carroll, was of Irish descent, granddaughter of Edward Carroll, who emigrated to Ohio in 1801. Both families had intermarried with Quakers and it was therefore natural that he should be sent to Haverford College, which was founded in 1833 by the Society of Friends.

Following his graduation from Haverford with an A.B. degree in 1876, Mr. Taylor entered the Class of 1877 at Harvard University. He took high rank in the work of the class and was considered the best cricket player in college. He had been devoted to his game at Haverford, where it is the leading sport, and continued to play it nearly to the end of his life.

After receiving his A.B. from Harvard in 1877, Mr. Taylor returned to Cincinnati, where he spent the next five years. He removed to Philadelphia in 1882 and for the next fifteen years was engaged in structural-steel and machine-shop work and sales engineering in connection with cranes, chain hoists, and similar machinery. In 1897 he became associated with the Westinghouse Electric & Manufacturing Co., at East Pittsburgh.

"He was concerned principally with commercial activities and sales as vice-president. He organized and systematized the sales department. He inaugurated salesmen's meetings at the factory in which sales representatives from different parts of the country came together for a week of conference and discussion. An important feature was the presentation by the engineers of the company of their recent products with explanation and discussion of construction and operating features. Mr. Taylor encouraged the salesmen to be self-reliant and to regard sales as a great game to be played with the intensity and enthusiasm with which he himself played cricket.

He took an interest in the welfare and opportunities for employees. He established the "Casino" near the works which comprised a restaurant, billiard parlors, and a night school. The latter has become the Westinghouse Technical Night School, an outstanding school of its type. Mr. Taylor took sympathetic and constructive interest in the development of the Westinghouse Club for young college men and its *Electric Journal*.

During his last four years with the company he was the senior and executive vice-president and was a member of the Board of Patent Control under which there was a joint use of patents by the General Electric and Westinghouse companies. The decade of his Westinghouse service was a prosperous one for the company—its

business increased tenfold and it became able to earn and pay 10 per cent dividends.

Mr. Taylor left the Westinghouse organization in 1906 and for about a year served as vice-president of the Yale & Towne Manufacturing Co., taking over administrative responsibilities at the Stamford, Conn., works.

Then, in 1909, he became general manager of Linotype and Machinery, Limited, of London. During six years with this company he traveled extensively in Europe, spending some time in Russia where he made a profound study of commercial conditions. Early in the World War he converted the linotype works at Manchester into a plant for national service, and he instituted a system of uniform scientific management in the English munition plants.

In 1915 Mr. Taylor was elected president of the S. S. White Dental Manufacturing Co., of Philadelphia, and he remained with that organization for more than ten years. He retired in 1926, but three years later he was back in active business as chairman of the executive committee of the Hall Electric Heating Company, Inc., an affiliated company of the General Electric. Later he served as chairman of the Liquidating Committee of the company.

Mr. Taylor was considered one of the most outstanding authorities in the United States on all subjects connected with foreign trade. From 1917 nearly to the end of his life, he rendered notable public service in connection with foreign commerce. He was a member of the National Foreign Trade Council, working for the welfare of overseas commerce, from 1917 to 1919. He became a director, in 1917, and later one of the vice-presidents of the American Manufacturers' Export Association, which he served until 1926, improving methods of the export business of this country and doing his best "to encourage the development of the international mind in the United States." He gave valuable service to the United States Chamber of Commerce from 1919 to 1926, first as a member of the Departmental Committee for Foreign Commerce, and secondly, as a member of a special committee to study and report on the adoption of the metric system in the United States. He was a member of the Advisory Committee on Foreign Trade of the United States Department of Commerce, under Secretary Hoover, and chairman of the Advisory Committee to the Philadelphia District Office of the Department. He also was a member of the Foreign Trade Committee of the Philadelphia Chamber of Commerce from 1920 to 1927, and was chairman the last year.

During the years 1917–1919 he was chairman of the Committee on Dental Instruments of the Medical Section of the Council of National Defense, and president of the War Emergency Association of Dental Manufacturers.

In 1926 Mr. Taylor was nominated by the Philadelphia Chamber of Commerce to serve on the Executive Committee of the Ocean Traffic Bureau of the Port of Philadelphia. He resigned in 1927 and during the next four years served as a member of Inter-American High Commission. He was assigned to the Committee for Uruguay and sat with this committee in both the third and fourth Pan-American Commercial Conferences.

Mr. Taylor had been a member of the A.S.M.E. since 1908 and was a former member of the American Institute of Electrical Engineers. He was a member of the Engineers' Club, New York, for many years, and one of its trustees from 1905 to 1909. He was also a member of the Harvard Club of New York, the Philadelphia Cricket Club, and the Racquet Club of Philadelphia, and was a founder of the Penn Athletic Club, Philadelphia. In 1930 he was awarded the Belmont Trophy by the Associated Cricket Clubs of Philadelphia, a silver plate inscribed "To Frank H. Taylor in recognition of his long and useful services to cricket extending over a period of more than fifty years." In 1934, at Haverford College commencement, he made an address and unveiled a tablet to William Carvill, who laid out and planted the Haverford Campus and who introduced cricket at the College.

In England he was a member of the London Chess Club, the Devonshire Club, and the Stoke Pogis Golf Club, and of several cricket clubs.

Mrs. Taylor relates the following incident: "When general manager and director of Linotype and Machinery, Ltd., of England, a strike seemed impending. He went up to the Manchester Works and organized a Cricket Club, joined it himself, and went out and played with the workmen. All were happy and the strike faded away. It was not the only time that he found a straight path to the heart of the Englishman through his fondness for cricket."

When the fiftieth anniversary of his marriage to Rebecca Nicholson was celebrated in 1930, Mr. Taylor wrote to his Class at Harvard: "All my family are living and well, viz.—the two old ones, four sons, their four wives, and thirteen grandchildren—a happy tribe of twenty-three."

Of his four sons—William N., F. Carroll, Norman H., and Roger W.—three were graduated from Harvard. Mr. Taylor said once that he always felt that his one year at Harvard was invaluable, introducing him into a new world, and he evidenced a warm interest in the university throughout his life.

Mr. Taylor died in Philadelphia on September 18, 1934. A notice published by *The Friend*, of Philadelphia, included the following tribute: "His personal charm and cheerfulness and friendly greeting made him beloved, and numerous young men have expressed gratitude for the inspiration of his example and advice. It is needless to say that his children and grandchildren gave him devoted love and admiration. During his business life, about two hundred thousand different people worked for him, and his simple note regarding this is: 'I enjoyed my work and they enjoyed theirs.' "

WILLIAM HENRY THOMFORDE (1881–1934)

William Henry Thomforde, associated with the Good Road Machinery Corporation, Kennett Square, Pa., from October, 1933, to February, 1934, when illness necessitated his going to a hospital, died on May 23, 1934. Surviving him are his widow, Charlotte M. (Starr) Thomforde, whom he married in 1902, and two children, Ernest Starr and Hilda Ruth Thomforde.

Mr. Thomforde was a recent member of the A.S.M.E., having joined the Society with the grade of member in 1932. He also belonged to the Masonic fraternity and Odd Fellows, being a past-master of the Beacon (N.Y.) Lodge of the latter, and was a former member of the Chelsea (N.Y.) Yacht Club.

Mr. Thomforde was born at Englewood, N.J., on June 27, 1881, son of Ernest A. and Nellie (Ruhling) Thomforde. He attended the grammar and high schools in Englewood and took a mechanical engineering course through the International Correspondence Schools.

After some years' experience as a draftsman on layouts for heating and ventilating systems for railroad stations, and power-plant, water-supply, and coaling-station layouts, Mr. Thomforde took a position in 1913 with the Dutchess Tool Company at Beacon, as designer of machinery for bakeries. He remained there until May, 1918, when he became chief draftsman and mechanical engineer in charge of the engineering department of the J. H. Williams Drop Forging Company, Brooklyn, N.Y. He designed dies, new tools and tool holders, wrenches, clamps, and gages for the company. During 1920 and 1921 he was chief draftsman for the Eisemann Magneto Corporation, Brooklyn, where his work included the design of magnetos and generators for use on automobiles and tractors of all kinds.

Early in 1922 Mr. Thomforde returned to the Dutchess Tool Company and for eight years was in charge of design and experimental and testing work there. From the beginning of 1930 until early in 1932 he was engaged in designing special machinery for making linoleum and cork products for the Armstrong Cork Company, Lancaster, Pa. A period of unemployment followed, prior to his connection with the Good Road Machinery Corporation.

ERWIN WILLIAM THOMPSON (1859–1935)

Erwin William Thompson, a pioneer in the development of the cotton and cottonseed-oil industry of the South, died at the home of his daughter, Francis Douglas (Mrs. C. D.) Lyle, College Park, Ga., on February 21, 1935. He was born in Colquit County, Georgia, on April 15, 1859, the son of William W. and Sarah (Graves) Thompson.

Mr. Thompson was graduated from Cornell University in 1881, and seven years later married Miss Eugenia Douglas Ladson of Cairo, Ga. He is survived by his wife and daughter, a grandson, D. C. Lyle, and three great-grandsons, Dan, Bruce, and Erwin Lyle.

While many of those interested in the cotton industry were content to engage in the production and sale of the fiber, Mr. Thompson extended his interest into that of cotton by-products. Soon after graduating from Cornell, he designed, built, and operated one of the early successful cottonseed-oil mills at Thomasville, Ga.

Out of this experience, he soon was recognized as an authority on cottonseed-oil production, and was called upon to design and manage other plants of a similar nature in the South. Included in these experiences was the management of the mill of the American Cotton Oil Company of Augusta, Ga., the construction and management of the Southern Cotton Oil Company at Montgomery, Ala., and later the management of other Southern Cotton Oil Company Mills at Houston, Tex., Columbia, S.C., and Charlotte, N.C.

After these years of experience, he was made auditor and construction engineer of the Southern Cotton Oil Company, with headquarters in New York, N.Y.

Subsequently, he served as manager of the textile department of Gregg and Company, New York, and as chief engineer with D. A. Tompkins Company, New York.

Throughout his active life, he was called upon repeatedly to design and build plants for the production of cottonseed oil. His close association with the industry brought him international recognition and

finally he was appointed to a post in the United States Department of Commerce in 1913, as an authority on cotton products. As a representative of the Department of Commerce he spent much of his time on special investigations in Europe and as a Commercial Attaché to American Legations and Embassies in Northern Europe.

From the many life experiences he gleaned the information which he carefully prepared for numerous contributions to the commercial and technical press and for three books—"Bookkeeping by Machinery," "Edible Oils in the Mediterranean District," and "Cotton Seed Products and Their Competitors in Northern Europe."

Mr. Thompson retired from active service in 1930 to be with his family and friends in Georgia. His membership in The American Society of Mechanical Engineers dated back to 1884, thus by the time of his passing he had been a part of the activity of the Society for fifty-one of the fifty-five years of its existence.—[Memorial prepared by Prof. W. R. Woolrich, Knoxville, Tenn., Mem. A.S.M.E.]

WILLIAM DAVID THORNHILL (1908–1934)

William David Thornhill, son of Mr. and Mrs. W. D. Thornhill, was born on February 24, 1908, at Bluefield, W.Va. He was graduated from the University of Virginia in 1932 with a B.S. degree in engineering. He served as vice-chairman of the A.S.M.E. Student Branch at the University in 1931–1932 and became a junior member of the Society following his graduation. He was also president of the Honor Committee of the University, vice-president of the engineering class of 1932, and Regent of the Pi Chapter of Theta Tau and a delegate to the 1931 convention of the fraternity.

At the time of his death on January 1, 1934, Mr. Thornhill was employed by the Lillybrook (W. Va.) Coal & Coke Co.

WILLIAM MYNN THORNTON (1851–1935)

William Mynn Thornton, whose death occurred on September 11, 1935, at Charlottesville, Va., was born in Cumberland County, near Farmville, Va., on October 28, 1851. His father, John Thruston Thornton, Lieutenant Colonel of the Third Virginia Cavalry, fell at the head of his regiment at Sharpsburg, September 17, 1862. His widow, who was born Martha Jane Riddle of Petersburg, Va., lived until May 1, 1897.

In his seventeenth year, 1868, William Thornton won his A.B. from Hampden-Sydney College, which was later to confer upon him the honorary degree of doctor of laws. He attended the University of Virginia from 1868 to 1870, and from 1871 to 1873, graduating in a number of schools without applying for a degree (a frequent choice at that time), during the first period, and pursuing special engineering studies in the second. He was as a student a member of the Phi Kappa Psi fraternity and of the Jefferson Literary Society. For later distinction, he was elected to the William and Mary Chapter of Phi Beta Kappa, and to the Raven Society of the University of Virginia, neither of which existed at the institutions of his student years.

In 1871–1873, he was assistant in mathematics and also, during the second session, in applied mathematics. Of his work at the University, he wrote that he took "all the mathematics then taught (pure, mixed, and applied), all the Natural Philosophy, all the Chemistry, except Analytical Chemistry, and had completed Latin under Peters, and Greek and German under Gildersleeve." He left the University in June, 1873, expecting to enter the engineering field. He found the gates closed and barred: a financial panic had swept over the country, and engineers everywhere were idle. He spent the whole summer hunting vainly for a job; there were no jobs, great or small.

In his emergency he was saved by an offer to teach at McCabe's University School in Petersburg. After completing his studies at the University of Virginia in 1873 he taught at the Bellevue High School in Bedford County for a year. He went into college teaching from Bellevue in 1874 when he was elected professor of Greek at Davidson College, North Carolina. After one session there, he was called to the University of Virginia to become adjunct professor, and in 1883 professor of applied mathematics and civil engineering. He performed the duties of that professorship until his retirement as professor emeritus in 1931. Professor Thornton was chairman of the faculty of the University of Virginia from 1888–1896, a position of importance in an institution that then had neither president nor dean. Under his direction the Engineering Department of the University was developed and expanded. It became a separate department after the election of a president for the University, in 1904. Professor Thornton became its dean and directed its policies and growth until his retirement. As a recognition of this important leadership, the engineering building bears the name of Thornton Hall.

Special honors came to him in 1900 when he went as United States Commissioner to the International Exposition in Paris, and in 1904 when he was a member of the jury of awards in civil engineering at the St. Louis Exposition. He was frequently a speaker on occasions of importance. Two of his Phi Beta Kappa addresses (one delivered before the mother chapter at William and Mary on The Old Humanities and The Humane Sciences) were published in the *Alumni Bulletin* of the University of Virginia to which he was often a contributor. Among the articles on biographical subjects in that periodical by Professor Thornton were sketches of John A. Broadus, John R. Thompson, Thomas R. Price, William H. McGuffey, George Frederick Holmes, John W. Daniel and Charles W. Kent. He contributed an article on Basil L. Gildersleeve to the *Library of Southern Literature*. His address, "Who Was Thomas Jefferson?" was published by the State Bar Association and in a separate copyright edition. He wrote several times on the Honor System of the University: The "Genesis of the Honor System," 1904, was printed in the *Marion Military Institute Bulletin* and "The Honor System at the University of Virginia in Origin and Use" in the *Sewanee Review*, January, 1907.

He was associated with Professor Ormond Stone in founding the *Annals of Mathematics* in 1884 and in editing the first twelve volumes. He was also a contributor to *Nouvelles Annales des Mathématiques*, *Van Nostrand's Magazine*, the *Railway Gazette*, and other publications.

Professor Thornton became a member of the A.S.M.E. in 1900 and was chairman of the Virginia Section of the Society in 1921–1924. He was a member of the Mathematical Association of America.

Dr. Thornton was married twice. His first wife, Eleanor Rosalie Harrison of the University of Virginia, whom he married in 1874 and who died in 1920, had six children: Dr. John Thruston Thornton of Wheeling, W.Va., Mrs. Eliza Carter Thurman of Charlottesville, Eleanor Rosalie Thornton of Boston, Janet Thornton of New York, Dr. William Mynn Thornton, Jr., of Johns Hopkins University, and Charles Edward Thornton of New York. He married Miss Gertrude Waller Massie in 1921.

Dean Thornton's distinction was threefold: As teacher and administrator in the Department of Engineering, as chairman of the faculty of the University of Virginia and leader in the general affairs of the University, and as a gentleman and scholar, unafraid—a speaker of force, a virile writer, and a rich personality.

The foundations for a school of engineering had been laid by Colonel Charles S. Venable, who had been assisted by Leopold J. Boeck, an able and idiosyncratic Pole who became assistant professor in 1867 and professor of applied mathematics in 1869. When on Professor Boeck's retirement, William M. Thornton succeeded him, he completely reorganized the course of study. "The first step," he wrote later, "in carrying out this policy was to discard systematic lectures and build up the best possible course on the basis of good textbooks. The second step was to prescribe the best available American Manuals as guides for the practical exercises of the students. The third was to select for engineering theory what was then regarded as the most comprehensive, accurate, and scientific treatise in the English language—Professor Rankine's Manuals. This course was followed without material modification for five years, until the desired tradition was firmly rooted in the life of the University. This end accomplished, a more flexible treatment was adopted." It was not until 1892, when in the basement of the Rotunda and Annex a new laboratory was installed with equipment, under Professor Thornton's direction, that he felt the University could teach engineering practice in all essentials. After the fire in 1895 and the consequent equipping of a new mechanical building, and more especially with the expansion that followed the election in 1904 of President Alderman, the engineering course was greatly developed through additions to the faculty and equipment. Dean Thornton's was the hand that guided this growth until just four years before the Department moved in 1935 into the splendid modern building to begin another period of expansion.

When Colonel Charles S. Venable was chairman of the faculty from 1876–1888, Professor Thornton was vice-chairman. In 1888 he succeeded to the chairmanship and was the active head of the faculty and University until 1896. Under these two administrators an active direction of the University's affairs led to a vigorous recovery from the postwar condition and to healthy growth in student attendance. It was a period marked by the Rotunda fire and the reconstruction that followed it. On Sunday morning at about ten o'clock October 27, 1895, the fire was discovered. In spite of heroic efforts the Annex was burned, the Rotunda wrecked, and about two thirds of the books in the library destroyed or lost. The Board of Visitors sent Dr. Thornton, chairman of the faculty, to New York and Boston to seek funds for a restoration. With Professor Echols, he served on the Visitors' Building Committee from 1896 to 1898. He secured the money with which the Rouss Physical Laboratory and the Randall Building were built, money for equipping the new mechanical laboratory, and generous donations from the Coolidge family of Boston and other friends of the University.

Professor Thornton was an exact and methodical teacher. His students remember him not only for his ready and tart wit but for the continuity which he preserved throughout his courses. He regularly reviewed the work of the preceding assignment and explained that of the next, while the day's assignment was demonstrated on the board. As a speaker, he was clear, incisive, and eloquent. He spoke, as he wrote, with a mellow grace combined with a hard brilliance that at times flashed out a truth with subtle ruthlessness. He belonged to a generation of Virginians who were not made in one mold and whose personalities were always at least as important as anything they did. They had distinction of manner and bearing, as well as of mind and achievement. They were dominating men, vivid and various in the manifestations of their individualities. Most of them were free from the qualities that attend self-exploitation and pompousness: but their presence was felt wherever they were, and a quiet dignity was more impressive, with its unconsciousness of self but expectation of due consideration, than any amount of conscious self-importance could have been. There is a story told of Dean Thornton's being called up once by a green official of an educational agency who wished to have him speak on some occasion. The telephone conversation was maladroitly begun, "Professor Thornton, I can use you next week—." "You can?" came the clear, crisp response. "I have never to my knowledge been used yet." It is not likely that in all his life he ever had been. He was a man who stood on his own feet and walked in no one's tracks.—[From a memorial prepared by James Southall Wilson, member of the faculty of the University of Virginia, for the University *Alumni News*.]

GEORGE TIPPETT (1873–1935)

George Tippett was born on June 23, 1873, at Cornwall, England, son of William James and Matilda (Binney) Tippett, and was educated in that country.

His first position in the United States was at the Mt. Hope (N.J.) mines of the Empire Steel & Iron Co., where he was general master mechanic from 1899 to 1907. During the next year he was master mechanic of the Belmont Tunnel, one of the tubes under the East River, connecting Manhattan with Long Island City.

In 1908 Mr. Tippett established his own business, the George Tippett Company, contractor for the installation of power plants and heating plants. With his office at Astoria, he became well known on Long Island for his exceptional ability in the field of heating. He contracted for installations in many large apartment houses on the Island and did some work in New York also.

Mr. Tippett became a member of the A.S.M.E. in 1923 and belonged to the Masonic fraternity. He was an able cornetist.

Struck by an automobile, Mr. Tippett was instantly killed at Devon, Conn., on November 22, 1935. He is survived by his widow, Louise (Nye) Tippett, whom he married in 1917, and by a daughter, Lucille.

ARTHUR JEAN TOWNSEND (1887–1935)

Arthur Jean Townsend, a notable contributor to the development of modern steel manufacture, died at the Harper Hospital in Detroit, Mich., on June 1, 1935. Since early in 1933 he had been vice-president and general manager of the Rotary Electric Steel Company, with whose president, H. M. Naugle, he had been associated since 1916 in the development of numerous processes and equipment used in the steel industry. Outstanding among these was the development of the wide stripsheet mills, the first installation of which was made at Butler, Pa., by the Columbia Steel Company of Elyria, Ohio. This installation was the forerunner of the present stripsheet mill development, one of the leading accomplishments in the steel industry to date.

Another achievement of Mr. Townsend and Mr. Naugle of particular importance was the rotary slitting process of producing ribbed expanded metal lath, which they worked out in the early part of 1920. This was the forerunner of the expanded beam section the patents on which were later turned over to the Jones & Laughlin Steel Co., and which is now being used extensively in the building industry.

Many other processes and machines were of importance in that they furthered the progress of the industry. Those on making semi-finished or finished steel products and on improvements in centrifugal molding apparatus and in making blooms, slabs, and billets were patented in many countries.

Mr. Townsend was born at Apollo, Pa., on October 5, 1887, son of Cort W. and Elsie (Kline) Townsend. He attended Ohio University for a time and then took the apprenticeship engineering course of the Westinghouse Electric & Manufacturing Co. From the time he completed this course in 1909 until 1916 he was connected successively with the Canadian Sheet Steel Corp., Ltd., Morrisburg, Ontario, Canada, the Portsmouth (Ohio) Steel Company, and the Berger Manufacturing Company, Canton, Ohio. His work for these companies related to the construction and operation of mills and power stations.

From 1916 to 1922 Mr. Townsend was vice-president and works manager of the National Pressed Steel Company, of Massillon, Ohio, and during the next two years held a similar position with the Columbia Steel Company. In the spring of 1924 he was made general manager of this company, and at the close of 1925 was further advanced to the office of vice-president and general manager of the Columbia Steel Company of Pennsylvania, also becoming vice-president of the Forged Steel Wheel Company of Pennsylvania. He retired in August, 1927, and for several years spent his time in travel. He entered business again in November, 1930, as vice-president and general manager of the newly organized Timken Holding & Development Co., with which he was connected until becoming associated with the Rotary Electric Steel Company in February, 1933.

Mr. Townsend became an associate-member of the A.S.M.E. in 1917, and belonged also to the American Institute of Electrical Engineers and to the American Iron and Steel Institute. His clubs included the Duquesne of Pittsburgh, Union of Cleveland, Detroit Athletic, Detroit-Canton Club, and Congress Lake Country and Red Run Golf Clubs. Golf was his favorite pastime. He helped a number of young men to secure a college education.

EDWARD GEORGE TREMAINE (1857–1935)

Edward George Tremaine, a member of the A.S.M.E. since 1889, was born in the Williamsburg section of Brooklyn, N.Y., on January 26, 1857, son of Dr. Edward Thomas Tremaine and Charlotte Eliza (Teare) Tremaine. At the age of seventeen he was apprenticed to A. T. Nichols & Co., of Williamsport, Pa. He was under instruction with the W. & A. Fletcher Co., New York, for six months and at the Pennsylvania Railroad locomotive shops at Renova, Pa., for two years. Completing his apprenticeship there in 1880 he returned to the Fletcher company in the position of journeyman machinist.

He supplemented his early public school education with evening courses in mechanical drafting at Cooper Union, New York, during the years 1881–1883, and attended the Vermont State Normal School at Castleton for a time.

In January, 1883, Mr. Tremaine began an association with P. Lorillard & Co., tobacco manufacturers, Jersey City, N.J., which continued until 1904 and again after 1912 until his death. Beginning as journeyman machinist, he rose in a few months to the position of foreman of the machine shop. He became assistant superintendent of design and construction and constructor of special machinery in 1886. In 1900 he also took the position of chief engineer, in charge of power generation and transmission.

Mr. Tremaine left the Lorillard company in 1904 and for eight years was engineer and designer of automatic machinery for the Automatic Weighing Machine Company, Newark, N.J. He returned to the Lorillard organization as general director and for many years designed power plants and their entire equipment, and branch factories in many parts of the United States. At the time of his death on December 25, 1935, he was serving the company as consulting engineer.

Mr. Tremaine was very much interested in philately and photography; the Brooklyn Institute of Arts and Sciences conferred a certificate on him for his work in the latter field.

Surviving Mr. Tremaine are his widow, Anna Wright (Linker) Tremaine, whom he married in 1890, and a son, Edward G. Tremaine, Jr., both of Wollaston, Mass.

JAMES C. TUCKER (1877–1935)

James C. Tucker, superintendent of the Masonic Home at Wallingford, Conn., died at Boston, Mass., on August 31, 1935, while on a vacation trip. His widow, Ida E. (Shappee) Tucker, and a son survive him.

Mr. Tucker was born at Jackson, Pa., on January 13, 1877. He was graduated from the State Teachers College at Bloomsburg, Pa., in 1898, and spent the next five years as a teacher and principal in public schools.

He began his engineering experience in 1903 as a machinist's helper at the Allis-Chalmers Company at Scranton, Pa. A year there was followed by a year in Cuba as superintendent of a manganese mining company, and a year as draftsman in the coal mining engineering department of the Delaware, Lackawanna & Western R.R. Co. From 1906 to 1910 he engaged in designing, drafting, and checking on steam turbines for the General Electric Company at Lynn, Mass.

Mr. Tucker returned to the teaching profession in 1910 as assistant principal of the Lynn English High School, supervising and teaching drafting, machine-shop practice, woodwork, and mathematics. He

continued in that position until 1917, when be became director of the State Trade School at South Manchester, Conn. Two years later he took the position of director of vocational education of the Department of Education of the City of Bethlehem, Pa. During four years there he developed and established industrial training courses in the public school system in coordination with the Bethlehem Steel Corporation and other industries in that city.

In 1923 Mr. Tucker was appointed general superintendent of prison industries, Department of Public Welfare, Commonwealth of Pennsylvania. He was in charge of all productive industries in the state penitentiaries and reformatories. During his five years in the office he moved and reestablished the automobile license tag industries; reorganized shoe, textile, and clothing industries; and established canning and furniture industries.

In 1928 Mr. Tucker became general sales manager for the Farnsworth Company, heat reclamation engineers, Conshohocken, Pa. In 1929 he accepted a position as superintendent of Plant No. 2 of the Carrier Engineering Company, at Newark, N.J., where he remained until May, 1931, when he went to Meriden, Conn., to become assistant superintendent of the Connecticut School for Boys. He assumed his duties as superintendent of the Masonic Home in January, 1934.

Mr. Tucker became a member of the A.S.M.E. in 1928. He was also a 32d degree Mason, a past-president of the Meriden Kiwanis Club, and a member of the Rotary Club of Wallingford.

ROBERT TIFFT TURNER (1886–1935)

Robert Tifft Turner, vice-president and general sales manager of Shepard Niles Crane & Hoist Corp., Montour Falls, N.Y., died in New York, N.Y., on November 13, 1935.

Mr. Turner was born in Elmira, N.Y., on August 5, 1886, the son of Robert Tifft and Helen E. (Boyd) Turner. He was graduated from the mechanical engineering course at Cornell University in 1908 and later attended the Graduate School of Business Administration at Harvard University for a year (1912–1913).

For a year after his graduation Mr. Turner was employed as a draftsman by the American-LaFrance Fire Engine Company, Elmira. He spent the summer and autumn of 1909 abroad, accompanying a mechanical engineer investigating European business practice.

In the spring of 1910 he became a special apprentice with the Niles-Bement-Pond Company at the Pond Tool Works, Plainfield, N.J. From there he was sent to the Bement-Miles Works in Philadelphia and later in the year to the Niles Crane Works in that city. In all three plants he was engaged largely in assembly work. In December, 1910, he was transferred again, to the Pratt & Whitney Co., at Hartford, Conn., where he worked first on assembly and later as a machinist in the production department, operating grinding and milling machines. In the fall of 1911 he demonstrated thread-milling machines and turret lathes in Connecticut and Western Massachusetts and then was appointed salesman for the company in the New York State territory.

After studying scientific management and accounting at Harvard during the winter of 1912–1913, Mr. Turner was located in the New York office of the Niles-Bement-Pond Company, as sales engineer for the foreign department.

Mr. Turner had been associated with the Shepard Niles Crane & Hoist Corp. since March 1, 1917, when he became sales manager in the New York office. He served in a similar capacity in the Philadelphia office from 1919 to 1921, when he was made sales promotion manager at Montour Falls. He was elected secretary and appointed general sales manager of the company in 1927 and two years later was elected vice-president.

A veteran of the World War, Mr. Turner saw service in France as a member of the 52nd Pioneer Infantry, which he entered as a private and in which he was later commissioned a second lieutenant.

Mr. Turner became an associate-member of the A.S.M.E. in 1916. He was a director of the Watkins State Bank and of the Watkins Glen Yacht Club. He belonged to the Harvard and Machinery Clubs in New York, and to several clubs in Elmira and vicinity. Surviving are two sisters, Mrs. H. H. Bickford and Miss Elizabeth U. Turner, and a brother, S. G. H. Turner, all of Elmira.

WARREN RUSSELL VALENTINE (1872–1935)

Warren Russell Valentine, mechanical engineer for the Pittsburgh Plate Glass Company, with which he had been connected since June, 1899, died at New Wilmington, Pa., on May 20, 1935.

Mr. Valentine was born at Ithaca, N.Y., on October 26, 1872, son of Warren and Mary (Russell) Valentine. He was graduated from Cornell University in 1894 with an M.E. degree in electrical engineering and for the next two years was director of the trade school and instructor in mechanical and freehand drawing at the New York State Reformatory at Elmira. He then entered the employ of E. C. Stearns & Co., manufacturers of bicycles and hardware, in Syracuse, N.Y., where he had charge of the selection, testing, inspection, and treatment of all material and finished parts of bicycles, and of the experimental department. Leaving there in 1898 he was connected with the American Steel Casting Company, at Alliance, Ohio, as superintendent until becoming associated with the Pittsburgh Plate Glass Company.

During the first part of his connection with the plate-glass industry, Mr. Valentine was assistant superintendent of Works No. 1 at Creighton, Pa. He laid out an installed electric equipment for changing from steam-engine to motor drive and was in charge of the reconstruction of the plant. He then was transferred in the same capacity to Works 3 and 4 at Ford City, Pa. In 1902 he was advanced to the position of superintendent of the Fuel, Steam, Power and Light Department of Works 3–4 and 5 and during the following year designed and installed an ash handling system, gravity oiling system, and a table for grinding and polishing plate glass. From 1903 until 1926 he served as superintendent of Shop No. 2 at Ford City. He designed and constructed the water-cooled casting table and roller in use in modern plate-glass plants, designed and constructed tables and runners for grinding and polishing, tables for cutting, frames and clamps for handling, and other plate-glass machinery. He took out patents on an annealing lehr, grinding runner, and on a process for briquetting cast-iron and steel borings. Since 1926 he had been located in the general office of the company at Pittsburgh.

Mr. Valentine became a member of the A.S.M.E. in 1916 and belonged to the American Society for Testing Materials. He was a Knight Templar and Shriner, and a member of the Cornell Alumni and the Keystone Club of Pittsburgh. He is survived by his widow, Alice (Ingram) Valentine.

RICHARD M. VAN GAASBEEK (1877–1935)

Richard M. Van Gaasbeek was born in Brooklyn, N.Y., on September 17, 1877, the son of Henry Reynolds and Emma Maria Van Gaasbeek. After attending public school in Brooklyn he served a five-year apprenticeship with John Lee's Sons in that city and was employed in bench and machine woodworking in New York and Elizabeth, N.J., until 1904. During the next four years he was foreman of the Burwell Building Company in Brooklyn and from then until 1914 carried on his own business as general contractor.

During these years of practical experience Mr. Van Gaasbeek was also taking correspondence and evening courses in contracting, carpentry and joinery, architectural drawing, and training of trade teachers. In 1914 he became an instructor in woodworking at Pratt Institute, Brooklyn. He conducted classes in both day and evening school and a few years later became head of the department of woodworking and supervisor of evening classes. He retired in June, 1935, following a leave of absence during the second semester of the year 1934–1935, and died on August 23, 1935.

Of Mr. Van Gaasbeek's years at Pratt Institute, S. S. Edmands, director of the School of Science and Technology, wrote:

"Mr. Van Gaasbeek was an exceptionally accomplished teacher of carpentry, millwork and joinery, serving Pratt Institute's School of Science and Technology in such capacity for twenty years, including fourteen years as head of the School's department of wood shop practice. Previously, he had become an expert carpenter and wood-millworker by trade. He carried into his teaching a remarkably wide and progressive knowledge of these subjects, and he displayed very notable ability to enlist the interest and enlarge the understanding of his students. He was especially interested and active in the development of the subject of wood technology."

Mr. Van Gaasbeek held a vocational teacher's certificate, granted by the University of the State of New York. After joining the teaching staff at Pratt Institute he took courses in the strength of materials as well as courses in English and German there, and a course in tests and measurements in the summer school at the State Normal School at Oswego, N.Y., where he was instructor in carpentry for several years. He was the author of several textbooks, including "A Practical Course in Wooden Boat and Ship Building," "A Practical Course in Roof Framing," and "A Course of Study of the House Carpenters Trade," and also contributed articles to trade magazines. He had served as expert examiner for carpenters or the Municipal Civil Service Commission, New York, N.Y.

Mr. Van Gaasbeek became an associate-member of the A.S.M.E. in 1930, and also belonged to the Society for the Promotion of Engineering Education and the New York Lumber Trade Association. He is survived by his widow, Emma M. (Adams) Van Gaasbeek, whom he married in 1902, and by a married daughter, Emma Ruth Goetcheus, and a granddaughter, Alma Lee Goetcheus.

WILLIAM FRANCIS WALSH (1882–1935)

William Francis Walsh, who died on March 19, 1935, in New York, N.Y., was born on October 19, 1882, in Cleveland, Ohio. His parents were James Francis and Mary Monica (Deiger) Walsh. He attended St. Patrick's Parochial and the West High schools, and St. Ignatius College, all located in Cleveland, and from 1900 to 1902 served a special apprenticeship and worked as assistant engineer of tests for the Norfolk & Western Ry., Roanoke, Va.

With the exception of a short time with American Steel Foundries in St. Louis and Chicago in 1906 in charge of the erection of exhibits and shop inspection, Mr. Walsh was located in Virginia until 1912. He was with the American Brake Shoe & Foundry Co. in 1903–1905; the Richmond, Fredericksburg & Potomac Ry. in 1907; and during the next four years with the Virginia Air Line Railway and its successor, the Chesapeake & Ohio Railway. He became general air-brake inspector and acting superintendent of motive power for this road in 1909 and his duties included general locomotive and car design.

In 1912 Mr. Walsh went to Chicago to enter the employ of the Galena-Signal Oil Company as mechanical expert in charge of the lubrication of locomotives and cars on the Chicago & Northwestern and Chicago, Milwaukee & St. Paul railways. He continued in this work until June, 1917, when he was commissioned a captain in the Corps of Engineers. He was ordered into active service in the fall of that year and was stationed at Camp Grant and Fort Leavenworth until January, 1918. He continued as a reserve officer until May, 1933.

After leaving active service, Mr. Walsh was in charge of shops and rolling equipment on the Chicago, Milwaukee & St. Paul Railway until July, 1919, when he became supervising service engineer of the Galena-Signal Oil Company in charge of lubricating engineers on all roads west of Chicago. In April, 1926, he was appointed district manager for the company at St. Louis, but resigned a few months later to enter the service of the Standard Oil Company of New Jersey, with headquarters at New York. He organized the Railroad Lubrication Department and served as manager of railway sales until his death. In 1932 the company presented him with an emblem for his twenty years of service in the field of railway lubrication. His thorough understanding of the problems of railway lubrication, combined with a pleasing personality, natural ability in the sales field, and extensive acquaintanceship in railroad circles, made him of great value to the company.

Mr. Walsh became an associate of the A.S.M.E. in 1915 and a member in 1923. He belonged to the International Railway Fuel Association, Traveling Engineers' Association, Brotherhood of Locomotive Engineers, New York Railroad Club, Racquet Club, Washington, D.C., and Downtown Athletic Club, New York.

As a hobby he raised blooded Percheron horses at his 600-acre estate in the Piedmont Hunt section of Loudoun County, Virginia.

Mr. Walsh married Kathryn Griffin in 1905 and is survived by her and by five children, Mrs. Harold Franklin Hall, and Mary M., William F., Jr., Dorothy D., and James A. Walsh.

FRANK OREN WELLS (1855–1935)

Frank Oren Wells, a leader in the development of the tap and die industry in Greenfield, Mass., died at his home there on June 23, 1935. He was vice-president of the Wells Tap & Die Co., which he organized in 1924, and owner of the Wells Manufacturing Company, which was formed in 1928, absorbing the Wells Tap & Die Co. He was also a director of the Storms Drop Forging Company, of Springfield, Mass., and president of the Granite State Mowing Machine Company, of Hinsdale, N.H.

Mr. Wells was born at Shelburne Falls, Mass., on January 6, 1855, the son of Elisha and Lucina (Lilly) Wells. After attending the Shelburne Falls local schools and Petersham Academy, he served a three-year apprenticeship with the Wiley & Russell Mfg. Co., Greenfield, for which his father was one of the first salesmen. At the age of 21 he joined his father and older brother, Frederick, in forming the Wells Brothers Company, which they developed gradually, working on a very limited capital, to a substantial business. Among their products was an improved form of die which soon developed into the present form of the "Little Giant" die, an innovation in the screw cutting tool industry which carried the name of the company throughout the world. In 1912 the company was merged with the Wiley & Russell Mfg. Co. to form the Greenfield Tap & Die Corp., one of the largest manufacturers of taps and dies in the world. He served this company for a number of years as president, resigning in 1919.

Mr. Wells was also president of the Weldon Hotel Company; The Weldon Hotel, in Greenfield, built under his direction in 1905, is widely known. He was greatly interested in agriculture and was president and a prime mover in the affairs of the Franklin County Agricultural Society for years. He was president of the Green River Cemetery Association and was largely responsible for great improvement in the appearance of the cemetery and its surroundings. He was a former director of the Franklin County Hospital Association, of Greenfield. He was a member of the Greenfield Club and the Hardware Club, New York, and a former member of the New York Engineers' Club, and several other clubs in Greenfield and Springfield, Mass. He belonged to the Second Congregational Church in Greenfield.

Mr. Wells became a member of the A.S.M.E. in 1903 and served the Society as a manager for the term 1918–1921.

When in June, 1912, the Society decided to supplement its "Standard Proportions for Machine Screws" which had been completed and published five years before (May, 1907), Mr. Wells was named as a member of the committee of nine on "Limits and Tolerances in Screw Thread Fits." The records indicate that it was the questionnaires and discussions, as well as the exigencies of the Great War, which led gradually to the suggestion that possibly the only way to bring about the unification of screw-thread practice in the United States would be by the appointment of a federal commission for this purpose.

A bill to create such a commission was offered to Congress by Congressman J. Q. Tilson, of Connecticut, in January, 1917. This first draft called for the representation of the Society by one member of the commission and Mr. Wells's name was included in a list of members considered for appointment should the bill be passed. Numerous hearings on this and subsequent drafts of the bill were held at which Mr. Wells was a frequent visitor. Finally, on July 18, 1918, a revised bill was passed by Congress.

The personnel of the National Screw Thread Commission was named by the Secretary of Commerce in July, 1918, and Messrs. James Hartness and F. O. Wells were given membership on the nomination of the A.S.M.E. Mr. Wells served as a member of the Commission until it was discharged in June, 1933.

The *American Machinist* once wrote of him. "In Washington he is known as Wells, the gage man; in The American Society of Mechanical Engineers as Wells, the screw thread man; in the New England Hotel Men's Association, as Wells, the hotel man; and among agriculturists as Wells, the farmer."

Mr. Wells was twice married. His first wife, Alice L. Graves, of Whately, Mass., whom he married in 1880, died in 1891, leaving a daughter, Dorothy Virginia, now Mrs. J. Tennyson Seller, of Greenfield. He is survived by her and by three grandchildren, and by his widow, Caroline (Dutton) Wells, whom he married in 1893.

WARREN MacCLELLAND WELLS (1884–1935)

Warren MacClelland Wells, insurance engineer for Cornwall & Stevens, with headquarters at Jackson, Miss., died on December 17, 1935, at Raleigh, N.C., while there on a business trip.

Mr. Wells had been associated with Cornwall & Stevens since December, 1926. His work consisted of making examinations, principally of cottonseed-oil mills and cotton gins, for the purpose of seeing that the plants were kept clean and that they were provided with sufficient fire-fighting appliances to insure adequate protection. His territory covered the states of Mississippi, Louisiana, Alabama, and Georgia. His work was highly regarded by the clients of the company.

The son of John and Cinda G. (MacClelland) Wells, he was born on April 30, 1884, at Norwood, N.Y., and secured his early education there. He was graduated from Syracuse University with a degree in mechanical engineering in 1907, and immediately took up the work of fire-prevention inspector. From 1907 to 1911 he was connected with the Middle States Inspection Bureau, New York, carrying on the inspection of manufacturing plants for some thirty-five insurance companies. During the next five years he was fire-prevention engineer and manager of the inspection department of J. S. Frelinghuysen & Co., New York, and for two years, beginning in June, 1916, handled fire-prevention work in dye, chemical, and ammunition plants as representative of Crum & Forster, New York.

In June, 1918, Mr. Wells was put in charge of fire protection and preventionat the U.S. Nitrate Plant No. 2 at Muscle Shoals, Ala., for the Air Nitrates Company. He remained there for nearly a year, the latter part of the time in charge also of safety engineering.

After leaving Muscle Shoals, Mr. Wells returned to general fire-prevention work, for the Pacific Fire Insurance Company, New York. He left in September, 1920, to become chief fire-prevention engineer for the General Motors Corporation, at Detroit, and for about two and a half years had charge of fire prevention and protection in all its plants. From the fall of 1925 until his connection in December, 1926, with Cornwall & Stevens, he was employed, first, by the Pull-

man Car & Manufacturing Co. in fire-prevention and safety work in its plants and the supervision of a million-dollar sprinkler and water mains installation at the plant at Chicago; and subsequently by the Mississippi State Rating Bureau, at Jackson, grading insurance rates for public water works and organizing and training a fire department.

Mr. Wells was a recognized expert in fire prevention and protection. He had a rating of 96 under the U.S. Civil Service at Muscle Shoals and was cited "for excellence" in his work there by the Air Nitrates Company and the Quartermasters Department of the U.S. Army. He had been an associate-member of the A.S.M.E. since 1921.

Mr. Wells was twice married and is survived by a son, Arthur M. Wells, by his first marriage, and by his widow, Ruby (Carroll) Wells, whom he married in 1919, and their two children, Rebel Carroll and Warren M. Wells, Jr.

WILLIAM WALLACE WHITE (1907–1934)

William Wallace White was killed in an airplane crash at Syracuse, N.Y., on April 2, 1934. He was the son of Henry Gowman and Pauline (Hammer) White of Seattle, Wash., where he was born on September 4, 1907.

Mr. White attended the University of Washington, receiving a B.S. degree in mechanical engineering in 1929. He was a member of the A.S.M.E. Student Branch at the University and coauthor, with Frank V. Bistrom, of a paper on "An Investigation of a Rotary Pump," which won the Student Award for 1929. This paper was published in Transactions, 1929, HYD-51-12. He was elected to membership in Sigma Xi, Tau Beta Pi, and Scabbard and Blade while attending the University.

During a number of summer vacations Mr. White assisted his father on building construction contracts, and he spent the summer of 1928 with a surveying party on the Diablo Dam for the City of Seattle. Since his graduation he had been with the General Electric Company for about three years, at Schenectady, N.Y., and then at Fort Wayne, Ind., where he worked in the Refrigerator Department until shortly before his death.

Mr. White became a junior member of the A.S.M.E. following his graduation from college in 1929. He enjoyed water sports, outdoor life, and music, and was a member of the Canoe Club and Mendelssohn Club in Schenectady. He was a first lieutenant in the Ordnance Reserve Corps., U.S.A.

FREDERICK OTTO WILLHOFFT (1876–1935)

Frederick Otto Willhofft, a former professor of mechanical engineering at Columbia University and at Queens University, Kingston, Ontario, Canada, died of pneumonia at the Lenox Hill Hospital in New York, N.Y., on February 6, 1935.

A native of Germany, Mr. Willhofft was born at Leipzig on March 14, 1876, son of Rudolph H. and Pauline Willhofft. He attended the Royal Gymnasium at Leipzig, securing the equivalent of a B.A. degree, and the Military College at Engers. After serving as an officer in the German army for several years, he came to the United States in 1898, and was first employed by a German architect in New York. Early in January, 1899, he entered the drafting room of E. W. Bliss Co., Brooklyn, where he worked until the fall of 1900. During the next four years he attended Columbia University, winning the Darling Medal and securing both A.M. and M.E. degrees.

The summer following his graduation was spent in the drafting room of the Gas Engine Department of the De La Vergne Machine Company, New York. Then he began a period of teaching which continued until 1917. The first two years were spent at the Clarkson Memorial School of Technology at Potsdam, N.Y., where he had full charge of the department of mechanical engineering, including the mechanical and testing laboratories. He began his work at Queen's University in 1906, organizing the mechanical engineering course, laboratories, and shops. During eight years spent in Kingston he also served as consulting engineer for the Canadian Locomotive Company and many other firms.

He returned to Columbia as assistant professor of mechanical engineering in 1914 and continued on the faculty for three years, also carrying on consulting work.

In addition to his consulting practice since 1917, Mr. Willhofft served from 1919 to 1923 as mechanical engineer for the Metal & Thermit Corp., New York, and secretary and treasurer of the Th. Goldschmidt Corp., 1923–1925, and had also been president and director of the S.A.M. Pump Corp., and secretary and general manager of the Isothermos Corporation of America. He became a naturalized citizen of the United States in 1927.

Mr. Willhofft was an associate of the A.S.M.E. from 1905 to 1919 and became a member in 1925. He also belonged to the Verein deutscher Ingenieure and was a former member of the American Institute of Electrical Engineers and of the American Society of Radio Engineers. He belonged to the Columbia University Club in New York. He is survived by his widow, Elsie (Miller) Willhofft, whom he married in 1930, and by a brother and two sisters in Germany.

ANTON P. WRIGHT, JR. (1901–1935)

Anton P. Wright, Jr., eldest son of Anton Pope and Hannah McCord (Smythe) Wright, died at the home of his parents in Savannah, Ga., on March 28, 1935, after an illness of five months of streptococcic septicemia. A descendant of prominent families of Georgia and South Carolina, he was born in Charleston, S.C., on October 1, 1901. His boyhood was spent in Savannah and he was graduated from the high school there in 1918. After attending the Georgia School of Technology for a year he entered Cornell University, from which he received an M.E. degree in 1923, and an E.E. degree four years later.

After his graduation in 1923, Mr. Wright became an apprentice engineer with the Dixie Construction Company, at Birmingham, Ala., a subsidiary of the Alabama Power Company. Subsequently he was transferred to Huntsville, Ala. With the exception of the year 1926–1927, when he had leave of absence for his studies at Cornell, he was in the employ of the Alabama Power Company until his death, at which time he held the position of assistant manager of the Montgomery District, with headquarters in Montgomery.

Mr. Wright became a junior member of the A.S.M.E. in 1924. He was past-president of the Junior Chamber of Commerce of Montgomery, a member of the Society of Colonial Wars, Phi Delta Theta fraternity, and the Country Club of Montgomery, and a communicant of the Episcopal Church of the Ascension of Montgomery. He was very fond of music and was interested in aviation. Two brothers who survive him are Augustin T. S. Wright and David Wright.

LAWRENCE ZAMBONI (1865–1933)

Lawrence Zamboni, who had retired from business and had been living in California for a number of years because of poor health, died in Pasadena on January 21, 1933. He had been a member of the A.S.M.E. since 1918 and also belonged to the Engineers Society of Pasadena and to the Masonic fraternity. He is survived by his widow, Rose (Drury) Zamboni, whom he married in 1892, and by two daughters, Marguerite (Mrs. Ernest S.) Brewster and Hilda Zamboni.

Mr. Zamboni was born at Modena, Italy, on January 9, 1865, son of Cataldo and Angelica (Bini) Zamboni. He attended the government technical school at Turin, Italy, completing his work there in 1889. From 1886 to 1889 he also worked on the design and construction of special machinery for the manufacture of military rifles at the government arsenal in Turin.

Following his graduation Mr. Zamboni went to London, England, where he was engaged for two years in the design and construction of quick-firing guns for the Maxim & Nordenfelt Gun & Ammunition Co. He then came to the United States, finding employment from 1893 to 1895 with the Standard Caster & Wheel Co., New York, N.Y.; his duties as superintendent of manufacturing included the design of dies and apparatus for the manufacture of sheet-metal articles. From 1896 to 1899 he was in charge of construction of special bicycles as superintendent of the Gomully & Jeffery Manufacturing Co., Brooklyn, N.Y. He became a naturalized citizen of the United States in 1899. During the next two years he was connected with the McCabe Hanger Manufacturing Co., New York, again working on the design and construction of tools and dies for the manufacture of sheet-metal articles.

Mr. Zamboni went to London in 1902 and until 1904 was in charge of the development department and superintendent of manufacture of the Scott-Snell Lamp Company. After returning to the United States he designed tools, dies, and apparatus for the manufacture of pressed-metal pulleys, wheels, and other sheet-metal articles for the American Pulley Company, Philadelphia, Pa.

From 1906 to 1912 Mr. Zamboni was factory manager and engineer of construction for the Bennett-Zamboni Manufacturing Co., New York, manufacturers of metal-working machinery, tools, jigs, and dies. He left to take charge of the laboratory established by the American Alloys Company, Newark, N.J., for the development of acid-resisting alloys. He also served as superintendent of manufacture for the company.

After 1917, until his health became poor, he maintained an office in New York as consulting mechanical engineer, devoting himself chiefly to the development of nonferrous acid-resisting alloys. During the World War he was called upon to serve as a supervising engineer in the Inspection Division of the Ordnance Department, U.S.A.

Mr. Zamboni had patented several inventions, including a pressed steel metal wheel, a coin carrier, and a skate roller.

Welding Design

By CHARLES H. JENNINGS,[1] EAST PITTSBURGH, PA.

Although welding has been extensively used in the fabrication of engineering structures for a number of years, there is still a decided lack of knowledge among engineers concerning the fundamental factors governing a satisfactory design. In addition, there is a lack of agreement among designers as to the methods of calculating weld stresses and the correct working stresses to employ for different types of joints. This article contains a discussion of a number of variables such as the selection of the proper joint, the calculation of weld stresses, the determination of working stresses and safety factors, and the important features governing a good welded design.

An analysis is made between butt and fillet welds in an effort to assist the designer in the selection of the proper type. Theoretical and practical aspects such as stress concentrations resulting from discontinuities in form, fabrication difficulties, welding costs, and distortion problems are considered.

The calculation of weld stresses in different types of joints is discussed and suitable formulas recommended on the basis of their agreement with test results, and their general applicability and acceptance in present design practice.

Working stresses and safety factors for butt and fillet welds are determined on the basis of static and dynamic tests. A table of recommended working stresses for bare and coated-electrode welds subjected to all types of loading is given to assist the designer. This table is based upon joints made on ordinary low-carbon structural steel.

A number of important design features that are essential in the design of economical welded structures are given. These features include the recommended minimum size of fillet welds for given plate thicknesses, the application of intermittent welds in design, and the influence of joint design on the economical fabrication of butt joints.

An appendix is attached which contains a number of typical welded-joint designs with the corresponding recommended formulas for calculating the stresses.

THE APPLICATION of welding to the fabrication of engineering structures and equipment presents a great many problems to the designer. Some of these problems are the result of the inherent properties of the deposited metal and the characteristic shape of certain welded joints while others are the result of the designers' efforts to create new and economical designs. Although many of the problems encountered are still unsolved and require a great deal of further research, the vast amount of data and experience which have been accumulated are sufficient to design rationally all types of welded structures and machines.

To design a welded structure properly it is important that the designer be thoroughly familiar with the following items: (1) Methods of calculating weld stresses, (2) allowable working stresses, (3) physical characteristics of parent and weld metals, (4) fabrication problems, and (5) inspection and testing facilities. A welded design may be satisfactory from the standpoint of strength but entirely unsatisfactory from the standpoint of materials and fabrication.

A careful analysis should be made of each structure to insure that it can be fabricated economically. Welds must be designed and located so as to keep distortion of the finished product to a minimum. Rigid joints should be eliminated as much as possible to prevent the development of excessive residual stresses which might cause cracked welds during fabrication or ultimate failure in service.

The choice between butt and fillet welds is subject to much controversy although when properly designed and fabricated each has definite advantages and each is entirely satisfactory. The choice of materials and filler material will vary greatly for different products and will depend largely upon the service conditions, the designer's knowledge of the weldability of materials, and the fabricating facilities available. In many cases two or more materials can be used for a given structure but the selection of the proper one will result in considerable saving in both fabrication and materials costs.

It is the author's intention to discuss how working stresses are determined for welds and to outline briefly the methods of calculating stresses resulting from static and dynamic loadings. In addition, a number of rules covering important points of weld design will be given.

No attempt will be made to discuss the problems of fabrication such as welding procedures, elimination of distortion, and choice of materials, although they are connected closely with the design of a successful structure.

Typical Welded Joints

In the design of welded structures there are two general types of welds used, butt welds and fillet welds (1).[2] These welds may be used in making many types of joints such as ordinary butt and fillet joints between parallel plates, T joints between plates joining each other at an angle, corner joints, and joggled joints. The proper selection between butt and fillet welds is of importance both from the standpoint of economics and the service life of the structure. Unfortunately, however, no set rule can be applied for selecting the proper weld.

Fillet welds in general require less preparation of the parts preparatory to welding because the parts may be lapped or butted together without the necessity of spending a great deal of time in beveling or preparing the plate edges. If the plates are lapped it is not essential that their dimensions be held to close tolerances.

[1] Engineer in charge of welding research, Westinghouse Research Laboratories. Jun. A.S.M.E. and Assoc-Mem. American Welding Society. Mr. Jennings was graduated from the mechanical engineering department of Iowa State College in 1928 with the degree of bachelor of science in mechanical engineering. He entered the graduate student course of the Westinghouse Electric & Manufacturing Company in 1928 and in 1929 became connected with the mechanics division of the research laboratories. In 1936 he was transferred to the chemical and metallurgical division of the research laboratories and appointed to his present position.

Contributed for presentation at the Welding-Practice Symposium sponsored jointly by The American Society of Mechanical Engineers and The American Welding Society and held at Cleveland, Ohio, October 22 and 23, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

[2] Numbers in parentheses refer to Bibliography at the end of the paper.

A variation of $^1/_8$ to $^1/_4$ in. in the amount of overlap of the plates will have no effect upon the strength of the joint providing the minimum requirements for overlap are maintained. Also if this variation is not the result of abrupt changes it will be impossible to detect it in the appearance of the completed structure.

In joints where the plates are butted at right angles to each other it is only necessary that the edge of the abutting plate be cut at right angles with the plate surface. This requires only a single cutting operation with the shear, cutting torch, or planer. The greatest problem encountered is to insure that the prepared edge is straight so that it will fit uniformly to the abutting plate. A space or gap between two abutting plates will reduce the effective size of the fillet welds and require that the weld size be increased by the amount of the gap. The increase in the size of a fillet weld caused by a gap between two plates may materially affect the amount of deposited metal required to make a weld of the required size. The percentage increase in the amount of weld metal required will be $[a(2h + a)100]/h^2$ where a represents the gap and h the required size of the weld. When making a $^1/_2$-in. weld between two plates, a $^1/_{16}$-in. gap represents an increase of 26.6 per cent in the amount of weld metal that must be deposited.

Butt welds in general require a better fit of the parts to be joined, and generally at least one of the butting edges is beveled[3] (thin plates excepted). Beveling of the plate edges is an added operation which must be considered in the cost of the structure.

The presence of a gap larger than necessary between the parts to be welded will also materially increase the amount of metal that must be deposited. Butt welds have an advantage over fillet welds in this respect however, because they are easier to inspect. After a fillet weld is made it is impossible to determine the presence of a gap between the parts by visual inspection; consequently, it is sometimes difficult to determine whether or not the weld is the correct size. This trouble is not encountered with butt welds because their particular design requires that the gap, if any, be entirely filled with weld metal.

The inherent shape of a fillet weld is such that it produces abrupt changes in the contour of the sections and consequently develops points of stress concentration. These stress concentrations are most severe at the root and toe (1) of the weld. Considerable theoretical work has been done on the investigation of fillet welds by photoelastic methods to determine the amount of stress concentrations. Solakian (2) found the stress at the root of a fillet to be six to eight times that of the average stress intensity in the connecting plates while the stress at the toe of the weld was three to five times the average stress intensity in the connecting plates. These stress concentrations varied with the external shape of the fillets and the amount of penetration and undercut present.

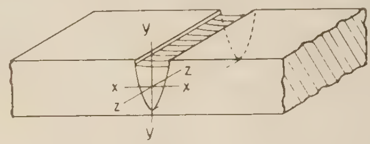
Butt welds in general have a more favorable form than fillet welds from the standpoint of irregularities which produce stress concentrations. A butt weld between parallel plates will produce no stress concentration providing it is a sound homogeneous weld and all the reinforcement has been removed. In actual practice, however, this condition is seldom obtained because nearly all butt welds have some reinforcement and a few internal flaws such as minute gas holes. Coker (3) found by photoelastic tests that the reinforcement on butt welds would produce a stress-concentration factor of 2.0. Small drilled holes representing flaws were also found to produce a stress-concentration factor of 2.0.

In cases where T joints are made between plates by using butt welds, there is the possibility of obtaining high stress-concentration factors at the junction of the plates. For fillet sizes of 0.05 in., the theoretical stress-concentration factor (4) will approach 2.5.

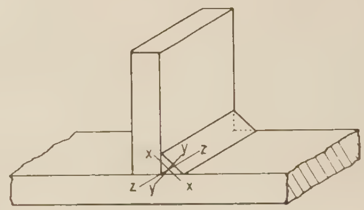
Under actual conditions, however, these stress concentrations are not as severe as the theoretical values might indicate. Under static loadings stress concentrations have little or no adverse effect upon the strength of a structure. The ultimate strength of the structure will not be lowered but the ability of the joint to deform plastically will be decreased somewhat, thereby tending to produce a more brittle type of fracture should failure occur. This apparent decrease in ductility does not weaken the structure.

In cases of dynamic loadings stress concentrations are of importance. Although from the data of Peterson and Wahl (4) it is evident that the actual reductions of fatigue strength for such cases of stress concentration as are applicable to welded joints are considerably less than photoelastic values.

This condition is true for holes and for surface irregularities. In the case of very small holes, the type generally obtained in welds, the reduction of fatigue strength was found to be small.



(a)



(b)

Fig. 1 Directions of Shrinkage When a Weld Cools

Fatigue tests on welded joints, as will be discussed later, have given similar results.

The fact that certain stress concentrations are present in welded joints should not be considered cause for preventing their use on structures subjected to dynamic loads. In the majority of welded structures subjected to dynamic loads, the complete loading consists of a combination of static and dynamic forces. The stress-concentration factor K resulting from the joint characteristics is applied only to the variable portion of the loading as shown in Fig. 17 and Equation [39], which will be discussed later. As a result, the increase in stress caused by the stress concentrations usually does not greatly increase the total stress on the joint.

From the standpoint of fabrication problems, two important points must be considered when selecting between butt and fillet welds.

Butt welds in general produce greater residual stresses. This fact has been proved experimentally and in production.

The reason for higher stresses being developed by butt welds is primarily the result of their characteristic shape. When a section of deposited metal solidifies and cools it tends to shrink uniformly in all directions. Referring to Fig. 1a, contraction tends to occur along the xx, yy, and zz axes. The contraction along the yy axis may be disregarded because the deposited metal is free and unrestrained at the top. The contraction along

[3] There are a number of welding processes being developed where beveling of the plates is not required. These processes are still under development and cannot be used in all types of structures, consequently, they will not be considered in this paper.

the xx axis tends to pull the plates together while the contraction along the zz axis tends to shorten the joint. Considering the contractions along the zz and xx axes, the latter is the more serious. This fact is readily realized when it is borne in mind that cracks in welds resulting from residual stresses are always parallel with the joint and not transverse as would be the case if the stresses along the zz axis were the more severe.

The degree of residual stress produced by the transverse contraction (along xx axis) is a function of the degree of fixity and the size of the plates. The reason such high stresses can be obtained is because this contraction produces a direct tensile force in the plates.

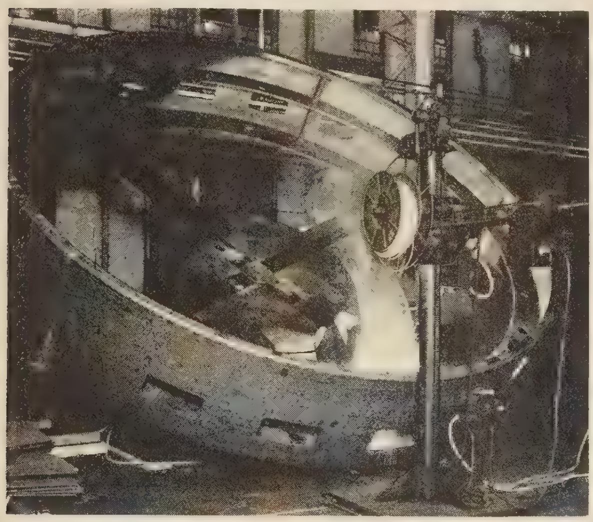


Fig. 2 A Generator Frame Positioned on a Welding Manipulator to Facilitate Down-Hand Welding

Considering fillet welds, shown in Fig. 1b, it is again possible to confine the discussion to contractions along the xx axis for reasons similar to those previously mentioned. In this case, however, there is generally the possibility of a small movement between the plates resulting from the weld contraction which will greatly reduce the stresses. In addition, this contraction is in such a direction that it tends to bend the parts rather than produce transverse tensile forces in them. As a result some local distortion is apt to be obtained and the presence of this distortion means a reduction in the residual stresses.

The second point in connection with the influence of fabrication problems on the choice between fillet and butt welds is the method of making the welds. In order to increase the speed of welding it is desirable to deposit the weld metal in the down-hand position with large-diameter electrodes. Butt welds are ideal in this connection and are generally preferred. Fillet welds are of such a nature that in the normal horizontal position one fusion zone is in the vertical plane. This necessitates the use of small-diameter electrodes, if welds of the highest quality are desired, unless the parts can be positioned so as to simulate a butt weld and permit down-hand welding. Welding manipulators as shown in Fig. 2 have been found very helpful under these conditions.

In cases where fillet welds of intermediate quality are satisfactory, special electrodes have been developed which make it possible to use $^1/_4$ in. diameters. Such cases make fillet welds as economical as butt welds.

Another factor in connection with the selection between butt and fillet welds is that butt welds allow the use of higher design stresses. This point will be discussed in detail later.

Reviewing the foregoing discussion, it is evident that both butt and fillet welds have definite advantages and the proper selection between them depends upon many factors. For a designer to create the most satisfactory and economical structure it is essential that all of these variables be carefully considered.

Calculation of Weld Stresses

The calculation of stresses in welds is of prime importance in connection with the design of every welded structure. Regardless of this fact, there is a surprising lack of agreement among authorities, particularly with reference to fillet welds, as to the proper methods of analysis. This lack of agreement may be attributed primarily to the characteristic shape of fillet welds and the many attempts to account theoretically for the nonsymmetrical stress distribution and the secondary bending moments encountered.

The object of this paper is not to give a highly theoretical analysis of the stresses in butt and fillet welds, but to discuss the commonly used methods and illustrate their application in the design of all types of structures and joints.

In the analysis of the following joints and connections the following notations will be used:

S = normal stress, lb per sq in.
S_s = unit shear, lb per sq in.
M = bending moment, in-lb
I = moment of inertia, inch units
K = stress-concentration factors
P = external load, lb
h = size of weld, in. For fillet welds h represents the weld leg and for butt welds h represents the throat of the weld excluding reinforcement
l = length of a weld, in.
L = linear distance, in.

Stresses in Butt Welds

The calculation of stresses in a butt weld between parallel plates as shown in Fig. 3 is a simple matter. The stress is equal

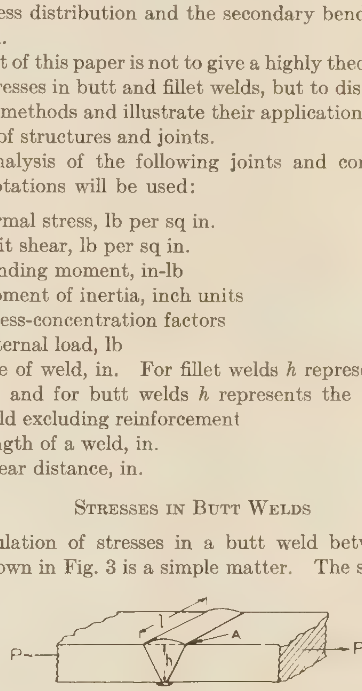


Fig. 3 Typical Butt Joint

to the external load acting on the joint divided by the throat area of the weld.[4]

$$S = P/hl \quad [1]$$

The stress in a butt weld due to shear loading is

$$S_s = P/hl \quad [2]$$

The values of h (weld throat) in Equations [1] and [2] do not include the reinforcement of the welds. Some authorities take the reinforcement into account but this is a questionable practice. The reinforcement of a weld will vary greatly over its length and is a maximum at the throat section. At the junction of the weld and the parent metal, shown as point A in Fig. 3, the reinforcement will approach zero, thereby making it the critical section. Also, reinforcement tends to produce stress concentrations which might be objectionable in cases of fatigue loadings.

The purpose of reinforcement on butt welds is to add an additional factor of safety to compensate for flaws which might be obtained when making the welds. As a result it is highly

[4] The weld throat is defined as the minimum thickness of a fusion weld along a straight line passing through its root (1).

desirable in many cases but it should never be used by the designer as a method of developing the strength required to withstand the applied loads.

Stresses in Fillet Welds

The stress distribution in fillet welds has been proved by photoelastic methods to be nonuniform (2, 3). In addition, their

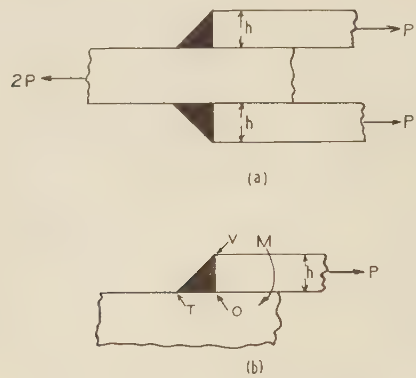


Fig. 4 Transverse Fillet Weld

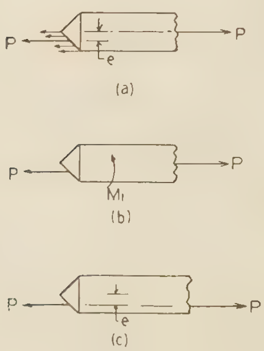


Fig. 5 Equilibrium of Forces in a Transverse Fillet Weld

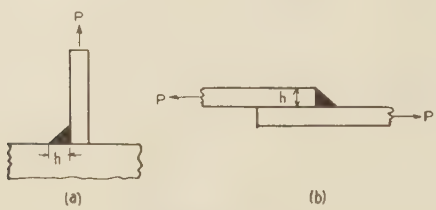


Fig. 6 Nonsymmetrical Fillet Welds

particular shape causes certain secondary bending moments which cannot be determined accurately and which tend to increase or decrease the stresses at different points in the weld. Because of these facts many attempts have been made to calculate fillet-weld stresses with the result that numerous formulas have been derived (5, 6, 7, 8). These formulas are all based upon arbitrary assumptions, consequently, none are strictly correct. For this reason only the generally accepted formulas will be considered.

Transverse Fillet Welds. In the generally accepted method of computing stresses in transverse fillet welds it is assumed that the stress at the throat section[5] is principally a normal tensile stress. (Photoelastic tests by Dustin (9) tend to justify this assumption.) The stress on the throat section of a fillet weld shown in Fig. 4 transmitting a load P is therefore taken as

$$S = \frac{P}{\text{weld throat}} = \frac{P}{0.707hl} = \frac{1.414P}{hl} \quad\quad [3]$$

where 0.707 = cosine of 45 deg.

If the force P acts through the center of the welded bar as shown in Fig. 4*b*, it will produce a bending moment M on the weld and the stress given by Equation [3] will not be the complete stress. On a joint of the type shown in Fig. 4*a*, however, a transverse force acts between the overlapping surfaces of the joint and this force produces some bending moment M_1 as indicated in Fig. 5*b* which acts in the opposite direction to the bending moment M caused by the force P. A shearing force also results from this reaction but it is probably small in comparison to the other forces and will be neglected. The force P and the moment M_1 shown in Fig. 5*b* can be represented by a force P eccentrically applied to the welded member as shown in Fig. 5*c*. The condition of equilibrium will be obtained when the bending moment M_1 is of such a value that the eccentricity e causes the load P to act along the center line of the weld throat. This is possibly the condition that occurs in a fillet weld at the point of failure. Consequently, the stress values obtained by this method are comparable to what would be expected from tests on all weld-metal test specimens.

There are certain cases in design when fillet welds are used on only one side of a plate as shown in Fig. 6. In joints of this type there is no counteracting bending moment present as the result

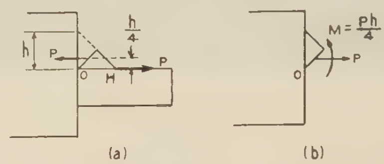


Fig. 7 Forces in Nonsymmetrical Transverse Fillet Weld

of the overlapping plates pressing against each other. Consequently, a bending moment resulting from the joint eccentricity must be considered.

Referring to Fig. 7*a*, it is seen that the weld throat is subjected to a tensile force P and a bending moment resulting from the eccentricity of the force P acting along the fusion zone OH. This bending moment M is equal to $Ph/4$.

The stress resulting from the direct load P is

$$S = \frac{1.414P}{hl} \quad\quad [4]$$

In determining the stress resulting from the bending moment M a rectangular stress distribution is assumed. (This is the type of stress distribution that is obtained before the weld fails in tension.) Here

$$S = \frac{4M}{(0.707h)^2l} = \frac{4Ph}{4(0.5)h^2l} = \frac{2P}{hl} \quad\quad [5]$$

This bending stress increases the stress at the root O and decreases the stress at the outer edge of the throat. As a result the root stress is the critical stress and the total stress is obtained by adding Equations [4] and [5]

$$S = \frac{1.414P}{hl} + \frac{2P}{hl} = \frac{3.414P}{hl} \quad\quad [6]$$

It is seen that for a given load P the stress obtained by Equation [6] is about 2.4 times the stress obtained by Equation [3].

[5] The throat section of a weld is considered to be the critical section.

Ultimate stresses in joints of the type shown in Fig. 6*a* as calculated by Equation [6] agree fairly well with results that would be expected from the tensile strength of the weld metal.

Parallel Fillet Welds. Parallel fillet welds are assumed to be subjected only to shearing stresses. The stress distribution along the weld is not uniform, although it is generally considered as such when calculating the stress. Smith (10) has shown experimentally that the stress at the ends of the welds is considerably higher than the average stress. (The exact value of the end stresses depends upon the length and size of weld and the size of the parts welded.) Weiskopf and Male (11) and Hovgaard (12) obtained similar results analytically.

Regardless of this fact, however, tensile tests made by Vogel (13), the U. S. Navy, and the Structural Steel Welding Committee of the American Bureau of Welding (14) indicate that the strength of such welds is directly proportional to the weld size and the weld length. As a result, the assumption that the stress is uniformly distributed over the entire length of the weld appears justified for normal design practice.

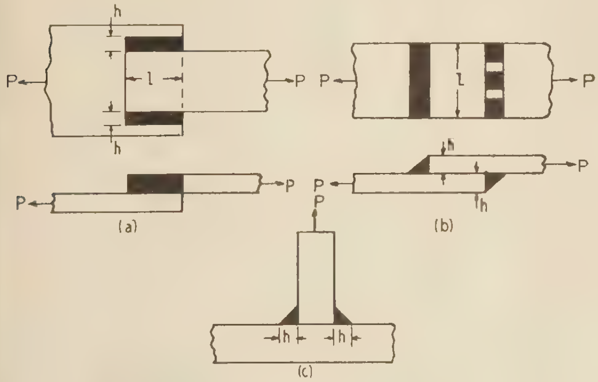


Fig. 8 Typical Fillet-Welded Joints

The shear stress in parallel fillet welds is calculated by dividing the load transmitted by the weld throat. This shear stress is

$$S_s = \frac{P}{\text{weld throat}} = \frac{1.414P}{hl} \qquad [7]$$

In cases where extremely long parallel welds are used, a formula based upon experimental constants has been derived by Rossell (15) which takes into account the stress distribution in the weld. In general welding design, however, there is no requirement for this formula, consequently, it will not be included.

Stress Calculations for Typical Connections

Equations [1] to [7], inclusive were derived for simple butt and fillet welds in tension and shear. In the following, typical joints subject to bending, tension, and shearing loads will be considered by applying these same equations.

When considering lap or T fillet-welded joints containing two parallel or transverse welds loaded in tension or compression as shown in Fig. 8, the load should be considered uniformly distributed between the welds. The weld stress is

$$S = \left(\frac{P}{2}\right)\frac{1.414}{hl} = \frac{0.707P}{hl} \qquad [8]$$

If plates of unequal thickness are used in combination with transverse fillet welds as shown in Fig. 9, the load distribution between the welds is proportional to the thickness of the plates because the plate sections between the welds do not distribute the load uniformly. Referring to Fig. 9, the stress in weld A is

$$S = \frac{a}{(b+a)}\,\frac{1.414P}{hl} \qquad [9]$$

and stress in weld B is

$$S = \frac{b}{(b+a)}\,\frac{1.414P}{hl} \qquad [10]$$

If side welds are used in making a lap joint between plates of unequal thickness, the load is considered uniformly distributed between the welds because the load distribution between the plates does not affect the load distribution on the welds. In this case Equation [8] applies.

Stresses in lap joints containing combinations of parallel and transverse fillet welds as shown in Fig. 10 are computed by assuming the load uniformly distributed between the welds. Al-

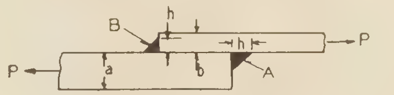


Fig. 9 Fillet-Welded Joint Between Plates of Unequal Thickness

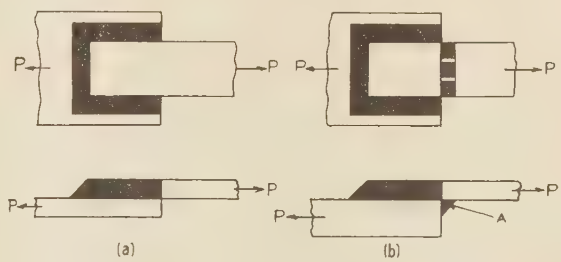


Fig. 10 Fillet-Welded Joints Containing Both Transverse and Parallel Welds

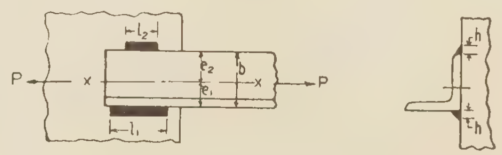


Fig. 11 Joint Between an Angle and a Plate

though this is the generally accepted method of calculating the stresses in joints of this type the load is actually not uniformly distributed between the welds. Tests have shown that parallel (side) welds deform more under load than transverse (end) welds. As a result the transverse welds will take more than their normal share of the load and will fail first. If the plates are of unequal thickness as shown in Fig. 10*b*, weld A will be the highest stressed. Although it is known that the load on joints of this type does not distribute uniformly between all the welds, there are not enough experimental data available to make a more accurate analysis.

Lap joints between an angle and another member are sometimes designed so that the center of gravity of the welds coincides with the center of gravity of the angle as shown in Fig. 11. Referring to Fig. 11, xx is the center of gravity of the angle, b is the width of the angle, and e_1 and e_2 are the distances from the center of gravity to sides of the angle. For the center of gravity of the welds to coincide with the center of gravity of the angle it is necessary that $l_1e_1 = l_2e_2$. Assuming the stress to be equal in both welds then

$$S = \frac{1.414P}{h(l_1 + l_2)} \qquad [11]$$

also

$$l_2 = \frac{l_1 e_1}{e_2} \qquad [12]$$

Substituting the value of l_2 in Equation [11] and simplifying

$$S = \frac{1.414Pe_2}{hl_1(e_2 + e_1)} = \frac{1.414Pe_2}{hl_1 b} \qquad [13]$$

Similarly

$$S = \frac{1.414Pe_1}{hl_2 b} \qquad [14]$$

The value of distributing the welds so that their center of gravity coincides with the center of gravity of the angle is open to question. Griffith (16) made comparative tests between joints of this design and joints having the weld uniformly divided, and proved one is equally as good as the other. Initial yielding occurred at approximately the same loads and the ultimate strength was approximately the same. On the basis of Griffith's tests and experience obtained on many structures, the author is of the opinion that it is unnecessary to design a joint on angle bars so that the centers of gravity of the angle and welds coincide.

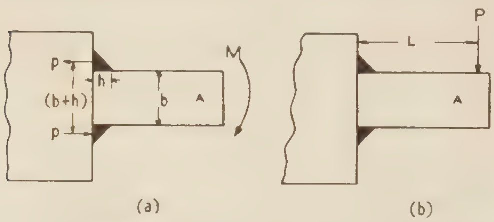


FIG. 12 TRANSVERSE FILLET WELDS LOADED IN BENDING

Transverse fillet-welded joints subjected to bending as shown in Fig. 12 are often found in machines. The weld stresses in this type of joint are computed by assuming the bending moment M to be counteracted by a couple composed of forces p acting at the center of the vertical fusion zones of the welds as shown in Fig. 12a. Therefore, $M = p(b + h)$, or

$$p = \frac{M}{(b + h)} \qquad [15]$$

The weld stress is

$$S = \frac{1.414p}{hl} = \frac{1.414M}{hl(b + h)} \qquad [16]$$

If a transverse load is applied to the member A as shown in Fig. 12b in place of a bending moment, the weld will be subjected to stresses resulting from the shearing force of the load P and the bending moment PL. Several methods have been suggested for combining these forces to determine the weld stress. The simplest and most commonly used method is given in the following paragraphs. This method is admittedly only an approximation but tests made on joints having a wide ratio of shearing and bending forces indicate that it gives satisfactory results (17).

Assuming both welds to be of equal length l, the shearing force on each weld will be $(P/2)$. From Equation [15], the force on each weld resulting from the bending moment PL will be

$$p = \frac{PL}{(b + h)}$$

The weld stress is computed by dividing the resultant of these two forces by the weld throat. Therefore, the weld stress is

$$S = \frac{1.414\sqrt{\left[\left(\frac{P}{2}\right)^2 + \left(\frac{PL}{b + h}\right)^2\right]}}{hl} \qquad [17]$$

$$S = \frac{P}{hl(b + h)}\sqrt{\left[2L^2 + \frac{(b + h)^2}{2}\right]} \qquad [18]$$

If the term $(b + h)$ is small in comparison to L, Equation [18] approaches Equation [16] for pure bending.

Parallel fillet-welded joints subjected to bending as shown in Fig. 13 are also commonly used. Stresses in this type of joint are calculated on the basis of the weld-throat sections' being subjected to the bending moment. The same formula is used in calculating the weld stress regardless of whether the bending is caused by a pure bending moment or a cantilever load. This, of course, is the result of the shearing stress being zero at the ends of the weld where the bending stresses are a maximum.

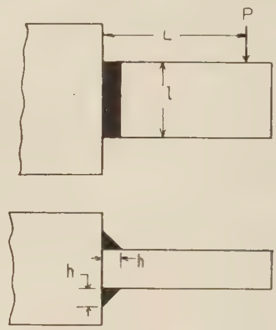


FIG. 13 LONGITUDINAL FILLET WELDS LOADED IN BENDING

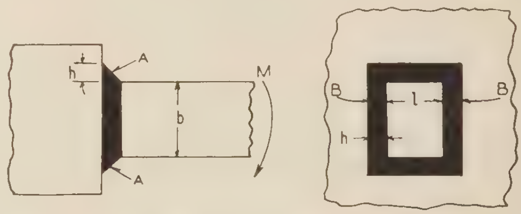


FIG. 14 FILLET-WELDED BAR SUBJECTED TO BENDING

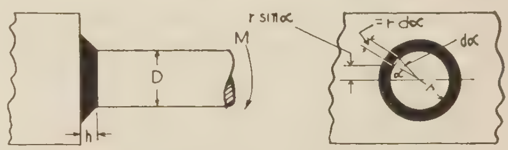


FIG. 15 FILLET-WELDED SHAFT SUBJECTED TO BENDING

Referring to Fig. 13, the bending moment acting on each weld is

$$M = PL/2 \qquad [19]$$

The section modulus of a weld throat is

$$\frac{0.707\, hl^2}{6} \qquad [20]$$

Therefore, the stress resulting from the bending moment is

$$S = \frac{6PL}{2(0.707)\, hl^2} = \frac{4.24\, PL}{hl^2} \qquad [21]$$

If a bar is fillet-welded to another member by welds on all sides as shown in Fig. 14 and subjected to a bending moment, the stress is calculated as follows:

The bending moment M_1 which is resisted by the transverse welds A can be calculated by Equation [16] where

$$S_1 = \frac{1.414\ M_1}{hl(b+h)} \quad \text{or} \quad M_1 = \frac{S_1hl\ (b+h)}{1.414} \ldots\ldots [22]$$

The bending moment M_2 which is resisted by the longitudinal welds B can be calculated by Equation [21] where

$$S_2 = \frac{4.24\ M_2}{hb^2} \quad \text{or} \quad M_2 = \frac{S_2hb^2}{4.24} \ldots\ldots [23]$$

If the maximum stress in all welds is the same, then $S_1 = S_2$ or $\dfrac{1.414\ M_1}{hl(b+h)} = \dfrac{4.24\ M_2}{hb^2}$

$$M_1 = M_2 \frac{3l(b+h)}{b^2} \ldots\ldots [24]$$

but $M = M_1 + M_2$. Therefore, by substituting in Equation [24],

$$M = M_2 \frac{3l(b+h)}{b^2} + M_2$$

$$M = M_2 \left[1 + \frac{3l(b+h)}{b^2} \right] \ldots\ldots [25]$$

Substituting Equation [23] in Equation [25] and solving for S

$$S = \frac{4.24\ M}{h[b^2 + 3l(b+h)]} \ldots\ldots [26]$$

The stress in a round bar, fillet-welded to another part and subjected to a bending moment as shown in Fig. 15, is calculated by assuming the stress in each weld element to be proportional to its distance from the neutral axis. It is also important in the derivation of the weld stress in this joint to bear in mind that the stress in the weld throat is (1/0.707) times the shear stress in the fusion zone of the weld, where 0.707 is the cosine of 45 deg.

Assuming an elementary area of the fusion zone $hr\ d\alpha$, shown in Fig. 15, on the periphery of the round bar at an angle α with the neutral axis subjected to a shearing force, df, the stress will be

$$ds = \frac{df}{hr\ d\alpha} \ldots\ldots [27]$$

Also, if S_s is the maximum shearing stress then

$$\frac{S_s}{r} = \frac{ds}{r \sin \alpha} \quad \text{or} \quad ds = S_s \sin \alpha \ldots\ldots [28]$$

Therefore, by substituting the value of ds in Equation [27]

$$df = S_s \sin \alpha\ hr\ d\alpha \ldots\ldots [29]$$

The bending moment developed by the force df is

$$dm = S_s hr^2 \sin^2 \alpha\ d\alpha \ldots\ldots [30]$$

By integrating Equation [30], the shearing stress S_s is obtained in terms of the bending moment M acting on the joint, or

$$S_s = \frac{M}{hr^2\pi} \ldots\ldots [31]$$

The stress in the weld throat as a function of the bending moment and the diameter of the welded bar will be

$$S = \frac{4M}{0.707hD^2\pi} = \frac{5.66\ M}{\pi hD^2} \ldots\ldots [32]$$

where 0.707 is the cosine of 45 deg.

If a welded joint of a design similar to that shown in Fig. 15 is subjected to torsion the weld stress is obtained as follows:

The torque M is resisted by the shearing force in the weld throat acting at a distance $D/2$ from the center of the joint

$$M = \text{shearing force} \times D/2 \ldots\ldots [33]$$

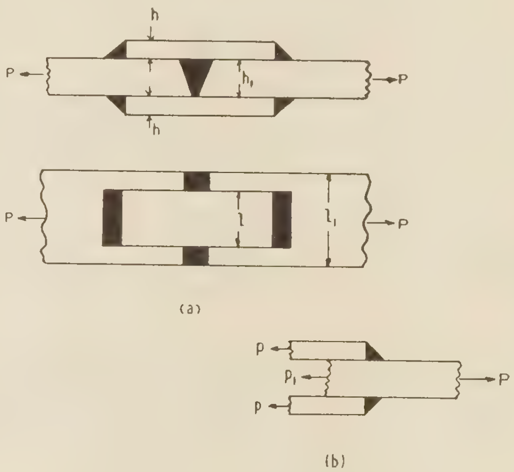


FIG. 16 REINFORCED BUTT JOINT

The shearing stress resulting from this shearing force is equal to the force divided by the weld throat. Therefore

$$S_s = \frac{2M}{0.707\pi D^2h} = \frac{2.83\ M}{hD^2\pi} \ldots\ldots [34]$$

When making a butt joint in a tension member it is often found necessary to reinforce the joint by an additional butt strap fillet-welded across the joint. This condition arises because the allowable design stress in a butt weld is not as high as the design stress for the parent material. If the parent material is stressed at 18,000 lb per sq in., and only 13,000 lb per sq in. is allowed on the butt weld, the entire member must be increased in size to bring the stress down to 13,000 lb per sq in., or a reinforcing bar must be added to carry the excess load. The latter of the two methods is usually the more economical.

When designing a joint of this type it is desirable to place a butt strap on both sides of the joint in order to eliminate secondary bending moments. This procedure, however, is not always practical. In such cases a bending moment is present which tends to increase the stress on the side of the butt weld away from the strap. If the butt weld is of the single V type it is desirable to have the root of the weld next to the strap.

The total elongation of all the plates that comprise the joint must be the same, assuming that the fillet welds do not deform. Consequently, the stresses in all the plates must be equal because the stress is directly proportional to the elongation. Also, if the stresses in the various plates are equal, the loads transmitted by the plates must be proportional to their cross-sectional areas.

If the load transmitted by each butt strap is p and the load transmitted by the butt-welded plate is p_1 as shown in Fig. 16b, then, because the load is proportional to the cross-sectional area of the plates

$$p = \frac{Phl}{(2hl + h_1l_1)} \ldots\ldots [35]$$

and

$$p_1 = \frac{Ph_1l_1}{(2hl + h_1l_1)} \qquad [36]$$

The stress in the fillet welds by using Equation [3] is

$$S = \frac{1.414Phl}{(2hl + h_1l_1)hl} = \frac{1.414P}{(2hl + h_1l_1)} \qquad [37]$$

The stress in the butt weld by using Equation [1] is

$$S = \frac{Ph_1l_1}{(2hl + h_1l_1)h_1l_1} = \frac{P}{(2hl + h_1l_1)} \qquad [38]$$

The welded joints that have been considered do not comprise all the joints that are used on welded structures. They do, however, cover the joints most commonly used, and the methods used in calculating the stresses can be applied in a similar manner to other types of joints.

Appendix 1 includes a number of typical welded joints with the corresponding formulas for calculating the stresses. These formulas are based upon the methods of stress calculation discussed.

Determination of Working Stresses

In the design of any welded joint it is important to know what safe working stresses can be used. The allowable working stresses are obtained from experimental tests by reducing the ultimate strengths or endurance limits obtained by a suitable factor of safety.

Many tests have been made on both fillet and butt welds. Consequently, it is not a difficult matter to obtain sufficient test data from which satisfactory design stresses may be determined.

There are two general types of welding electrodes used in the fabrication of welded structure; bare electrodes and coated or shielded-arc electrodes. Typical physical properties of weld metal deposited by bare and coated electrodes are given in Table 1.

Extensive tests on butt-welded joints indicate that the physical properties of butt welds are comparable to the physical properties of all weld metal. This condition is true regardless of the joint design, providing the weld is sound and homogeneous.

The working stresses for butt welds made with bare electrodes as recommended by the American Welding Society in the Structural Steel Welding Committee (18) and machinery construction codes (19) are given in Table 2.

Comparing the allowable stresses for static tension and compression given in Table 2 with the minimum ultimate strength of weld metal deposited with bare electrodes given in Table 1, it is seen that a safety factor of 3.4 is used in tension and 2.5 in compression. It is the author's opinion that the safety factor in

TABLE 1 PHYSICAL PROPERTIES OF METAL DEPOSITED BY BARE AND COATED ELECTRODES

Property	Bare electrode Min	Bare electrode Max	Coated electrode Min	Coated electrode Max
Yield point, lb per sq in.	35000	40000	42000	55000
Ultimate strength, lb per sq in.	45000	55000	60000	70000
Elongation in 2 in., per cent	8	15	25	35
Reduction of area, per cent	15	20	45	65
Endurance limit, lb per sq in.	16000	20000	26000	30000
Impact strength, Izod, ft-lb	5	15	40	50
Density, g per cc	7.5	7.6	7.81	7.85

TABLE 2 RECOMMENDED WORKING STRESSES FOR BUTT WELDS MADE WITH BARE ELECTRODES

	Structural code	Machinery code
Static tension, lb per sq in.	13000	13000
Static compression, lb per sq in.	18000	18000
Static shear, lb per sq in.	11300	11300
Dynamic tension, lb per sq in.	...	5100
Dynamic compression, lb per sq in.	...	7000
Dynamic shear, lb per sq in.	...	3400

compression is too small. The design stress for bare-electrode welds subjected to compression should be 15,000 lb per sq in. This represents a safety factor of 3.

The working stress for butt welds in static shear is 11,300 lb per sq in. This is 87 per cent of the working stress for static tension and may be questioned.

Tests made on butt welds in shear by the Westinghouse Electric and Manufacturing Company, the Structural Steel Welding Committee of the American Bureau of Welding (14) and the U. S. Navy indicate that the shearing strength of butt welds is about 65 per cent of the tensile strength. This ratio of shearing strength to tensile agrees closely to the theoretical ratio of 58 per cent as obtained by the shear-energy theory (20, 21).

In order to have the working stress in shear in agreement with the other values it should be approximately 60 per cent of the working stress in tension or 8000 lb per sq in. (8000 lb per sq in. is 61.8 per cent of 13,000 lb per sq in.).

The allowable working stress for butt welds subjected to dynamic loads should be the same for both tension and compression. For complete reversal of load, the design stress can be obtained from the endurance limit of butt joints or weld metal. Using 16,000 lb per sq in., which is a typical fatigue value for butt welds, and a safety factor 3.2, a working stress of 5000 lb per sq in. is obtained. This is the value used by the Westinghouse Electric and Manufacturing Company and agrees very well with that recommended by the A.W.S. machinery code (19). For dynamic shearing stresses a value about 60 per cent of 5000 should be used, or 3000 lb per sq in. This value is about 12 per cent lower than that recommended by the A.W.S. machinery code (19).

Considering coated or shielded-arc electrode welds, it is seen from Table 1 that the tensile strength is from 27 to 30 per cent greater than the tensile strength of bare-electrode welds. Therefore, it is satisfactory to increase the working stresses for static tension, compression, and shear to 16,000, 18,000, and 10,000 lb per sq in., respectively. The percentage increase in all cases is 25 per cent or less.

For dynamic loadings with this type of electrode using the minimum value of 26,000 lb per sq in. and a safety factor of 3.25, a working stress of 8000 lb per sq in. is obtained for tension and compression. The working stress for shear loadings can be established at 5000 lb per sq in. (62.5 per cent of 8000 lb per sq in.).

The tensile strength of fillet welds varies depending upon the design of the joint because of the type of loading, secondary bending moments and the approximate formulas used in calculating the weld stresses. In a report published by the Structural Steel Welding Committee of the American Bureau of Welding (14) it was found that side (parallel) fillet welds were approximately 25 per cent weaker than end (transverse) fillet welds. This is, of course, what would be expected because side welds are stressed in shear while end welds are stressed principally in tension. Also joints containing some eccentricity were found to be weaker than symmetrical joints.

In determining the working stress it has been the practice to use the weaker joints as the criterion, thereby having only one working stress for all types of fillet welds.

The working stress recommended by the Structural Steel Welding Committee (14) and the A.W.S. machinery code (19) is 11,300 lb per sq in. This value is derived on the basis of a tensile strength of 42,000 lb per sq in. and a safety factor of 3.7 (the stress of 42,000 lb per sq in. is the weighted average of some 761 specimens tested by the Structural Steel Welding Committee of the American Bureau of Welding).

This working stress appears to be entirely satisfactory and has been used for several years with good success. It is also very

convenient because it represents a load-carrying capacity of 1000 lb per linear in. per $^1/_8$ in. of weld. In other words, a $^1/_8$-in. weld 1 in. long will carry a 1000-lb load; a $^3/_8$-in. weld 1 in. long will carry a 3000-lb load and so on.

The fact that welds made with coated electrodes are from 27 to 30 per cent stronger than those made with bare electrodes makes it possible to increase their working stress to 14,000 lb per sq in. (This value is 24 per cent greater than 11,300 lb per sq in.) This working stress is also convenient to employ because it is approximately equivalent to the decimal equivalent of the weld times a factor of 10,000. A $^1/_8$-in. weld 1 in. long will carry 1250 lb; $^1/_2$-in. weld 1 in. long will carry 5000 lb, and so on.

The establishment of working stresses for fillet welds subjected to dynamic loadings is a difficult problem because of the large variation in the results which have been obtained. Also, in many cases the results given are based upon failure in the parent metal at the end of the weld and not in the weld metal. As a result they do not give fatigue values of the weld. Fatigue data on parent-metal failures at the ends of welds are important, however, in determining the stress-concentration factor at that point in the joint.

Fatigue tests made by the author and R. E. Peterson (22) on transverse fillet welds (bare electrodes) by means of rotating specimens, gave endurance limits of the 13,000 to 17,000 lb per sq in. for the weld. These tests also gave endurance limits for the parent metal at the weld toe of 18,000 lb per sq in. Kommerell (23) obtained endurance limits of 9200 to 14,600 lb per sq in. Hankins (24) obtained endurance limits of 9000 lb per sq in. for bare-electrode welds and 16,000 lb per sq in. for coated-electrode welds, while Roberts (25) obtained endurance limits of 17,900 lb per sq in. for bare electrodes. Ros and Eichinger (26) obtained values of about 18,000 lb per sq in. (computed by the author on the basis of recommended formulas) on coated electrodes.

On the basis of these data a working stress of 3000 lb per sq in. for transverse fillet welds made with bare electrodes and 5000 lb per sq in. for transverse fillet welds made with coated electrodes appears satisfactory. The fatigue strengths of parallel fillet welds are slightly lower than those of transverse fillet welds. Hankins (24) found them to be about 13 per cent less, Ros and Eichinger (26) about 19 per cent less and Roberts (25) about 25 per cent less. Kommerell (23), however, found parallel welds to be about 20 per cent stronger than transverse welds. Because of the large safety factor used for transverse fillet welds it is satisfactory to use the same design stress for longitudinal fillet welds. This greatly simplifies the design of fillet-welded joints and is in agreement with the method employed in selecting the static working stress for fillet welds.

Because the parent metal is weakened at the end or toe of a weld as a result of metallurgical changes and stress concentrations, it is desirable to determine the stress-concentration factor at these points. The stress-concentration factors given are based upon mild-steel parent metal. In cases where medium carbon or alloy steels are used, these stress-concentration factors may be greater because of more serious metallurgical changes.

Fatigue tests made by the author and the authorities mentioned indicate that the endurance limit of mild steel is reduced to about 18,000 lb per sq in. (22) at the toe of a transverse fillet weld. This corresponds to a stress-concentration factor of 1.5 (the endurance limit of mild steel is approximately 27,000 lb per sq in.). This factor appears to be the same for welds made with both bare and coated electrodes.

The endurance limit at the end of parallel fillet welds is 10,000 to 13,000 lb per sq in. This corresponds to a stress-concentration factor of 2.7.

TABLE 3 WORKING STRESSES AND STRESS-CONCENTRATION FACTORS FOR WELDS ON LOW-CARBON STEELS

Type of weld	Working stresses—bare electrodes: Static loads, lb per sq in.	Working stresses—bare electrodes: Dynamic loads, lb per sq in.	Coated electrodes: Static loads, lb per sq in.	Coated electrodes: Dynamic loads, lb per sq in.
Butt welds:				
Tension..........	13000	5000	16000	8000
Compression......	15000	5000	18000	8000
Shear............	8000	3000	10000	5000
Fillet welds:				
Transverse and parallel welds...	11300	3000	14000	5000

Stress-concentration factors

Location	Stress-concentration factor, K
Reinforced butt welds..........	1.2
Toe of transverse fillet weld..........	1.5
End of parallel fillet weld..........	2.7
T butt joint with sharp corners..........	2.0

On reinforced butt welds Roberts (25) obtained stress-concentration factors of 1.2 as compared to machined butt welds.

For T joints made with butt welds and having a right-angle corner, a stress-concentration factor of 2.0 will be found satisfactory.

The working stresses and stress-concentration factors that have been derived are given in Table 3. The stress-concentration factors given are applicable for welds made with both bare and coated electrodes, and need only be used in cases where dynamic loads are encountered.

Combined Static and Dynamic Loads

The design of structural members subjected to simple static or dynamic (complete reversal of stress) loadings is an easy matter on the basis of the previous design stresses given. In most structures this simple condition is seldom encountered, however, because most members are subjected to various combinations of static and dynamic loadings.

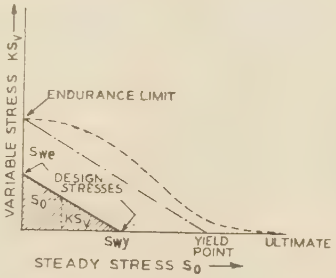


FIG. 17 RELATION BETWEEN STEADY AND VARIABLE STRESS

The accumulation of experimental data obtained on specimens under various combinations of static and dynamic loadings has made it possible to determine certain relationships. If the steady stress is plotted as abscissas against the variable stress as ordinates, the experimental curve will take the shape shown by the dotted line in Fig. 17. Gerber approximated this curve as a parabola while Goodman approximated it as a straight line between the endurance limit and the ultimate stress. It will be noted, however that the curve dips toward the yield point. Therefore, it is more simple and conservative to connect the endurance limit with the yield point as shown by the dot-and-dash line in Fig. 17 (27).

The line connecting the yield point and the endurance limit is assumed to represent the combinations of steady and variable stresses that will cause failure. It is necessary in design practice, however, to employ a suitable factor of safety. Consequently, the working stresses for variable and steady loads are used in

place of the endurance limit and yield point, respectively. This confines the permissible stresses within the shaded area of Fig. 17.

The relation between the steady stress S_o and the variable stress S_v as defined by the straight line joining the variable working stress S_{we} and the steady working stress S_{wy} (28) is

$$\frac{KS_v}{S_{we}} + \frac{S_o}{S_{wy}} = 1 \qquad [39]$$

The steady stress S_o is equal to one half the algebraic sum of the maximum and minimum stresses applied to the member or joint. The variable stress S_v is equal to one half the algebraic difference between the maximum and minimum stresses and the factor K is the stress-concentration factor for dynamic loads. If the maximum stress is S_{max} and the minimum stress is S_{min} then

$$\frac{K(S_{max} - S_{min})}{2S_{we}} + \frac{(S_{max} + S_{min})}{2S_{wy}} = 1$$

and

$$S_{max}(S_{we} + K\,S_{wy}) + S_{min}(S_{we} - K\,S_{wy}) = 2S_{we}S_{wy} \ldots [40]$$

If the loading of a joint is known, the weld size can be calculated by substituting in Equation [40] the proper expressions for S_{max} and S_{min} in terms of the loads and the weld size.

To illustrate the application of Equation [40], consider a butt weld between 1-in. plates. This weld is made with coated electrodes, the reinforcement is not removed, and the tensile load on the joint varies from 10,000 to 40,000 lb. The required length of the weld is determined as follows:

From Equation [1], $S = (P/hl)$. Therefore, by substituting the value of S in Equation [40]

$$\frac{P_{max}}{hl}(S_{we} + KS_{wy}) + \frac{P_{min}}{hl}(S_{we} - K\,S_{wy}) = 2\,S_{we}\,S_{wy}$$

$$l = \frac{P_{max}(S_{we} + K\,S_{wy})}{2h\,S_{we}\,S_{wy}} + \frac{P_{min}(S_{we} - K\,S_{wy})}{2h\,S_{we}\,S_{wy}}$$

From Table 3, $S_{we} = 8000$ lb per sq in., $S_{wy} = 16{,}000$ lb per sq in., and $K = 1.2$. Solving for l

$$l = \frac{40{,}000[8000 + (1.2 \times 16{,}000)]}{2 \times 1 \times 16{,}000 \times 8000} + \frac{10{,}000\,[8000 - (1.2 \times 16{,}000)]}{2 \times 1 \times 16{,}000 \times 8000}$$

$l = 4.2 - 0.44 = 3.76$ or 3.75 in.

If the load on this joint varied from 40,000 lb tension to 40,000 lb compression (complete reversal of load) the required length would be

$$l = \frac{40{,}000[8000 + (1.2 \times 16{,}000)]}{2 \times 1 \times 16{,}000 \times 8000} - \frac{40{,}000\,(8000 - (1.2 \times 16{,}000)]}{2 \times 1 \times 16{,}000 \times 8000}$$

$l = 4.25 + 1.75 = 6$ in.

If a welded joint were subjected to shearing stresses, Equation [40] still applies but the working stresses for shear are used.

General Design Notes

The design of a welded structure does not consist of simply designing the many joints to withstand the necessary loads. There are many economic and fabrication problems which must be considered in order to make the most satisfactory structure.

TABLE 4 MINIMUM-SIZE FILLET WELDS FOR DIFFERENT THICKNESSES OF PLATE

Plate thickness, in.	Minimum weld size, in.
1/8 to 3/16, inclusive	1/8
1/4 to 5/16, inclusive	3/16
3/8 to 5/8, inclusive	1/4
3/4 to 1, inclusive	3/8
1 1/8 to 1 3/8, inclusive	1/2
Above 1 1/2	3/4

In some cases these problems may be of sufficient importance to make the fabrication of the structure entirely impractical.

The ideal welded structure is composed of the fewest parts possible joined with the minimum amount of weld metal that is adequate for fabrication and service requirements. Whenever possible flanges and adjacent members should be bent from the same plate to eliminate corner welds. Structural steel plates and shapes cost about 2¢ a pound while deposited weld metal costs about $0.50 to $1 a pound. Consequently, the advantage of reducing the amount of welding is readily recognized.

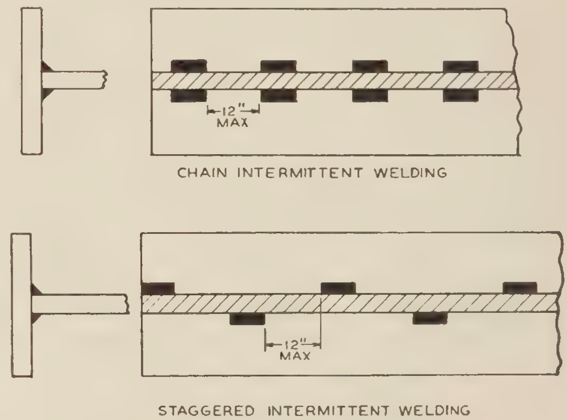


Fig. 18 Types of Intermittent Welding

When butt welds are used, the plate edges need not be beveled for thicknesses 1/4 in. or less. The edges of heavier plates however should be beveled to form some type of V joint. The best design of the joint; that is, whether it is a single or double bevel, single or double V, single or double U, or single or double J will depend upon a number of factors.

Oxyacetylene and oxyhydrogen cutting, is, in general, the cheapest method of preparing bevels for butt joints. It is adaptable to complicated shapes and suitable only for cutting plane kerf surfaces. Machining is particularly adapted for U-type joints, for cases where an excellent fit is required, and for parts of such a nature that they can be machined at a relatively low cost.

Double U- and V-type joints are recommended for plates 3/4 in. thick and over if it is possible to weld from both sides of the plate. This type of joint produces less distortion of the welded parts and reduces the amount of weld metal necesary to weld a plate of a given thickness.

The U-type joint with its rounded bottom makes it possible to make the first passes with an electrode of any desired diameter. The V-type joint is generally narrow at the bottom, consequently, the first passes must be made with small-diameter electrodes. Regardless of this fact, however, experience indicates that on plate thicknesses up to 1 in. there is little or no difference between the welding speeds obtained on the two types of joints.

The width of the bottom of a U-type joint greatly influences the welding cost. On plates up to 1 in. in thickness, it is advantageous to design the joint so that the first passes can be made with

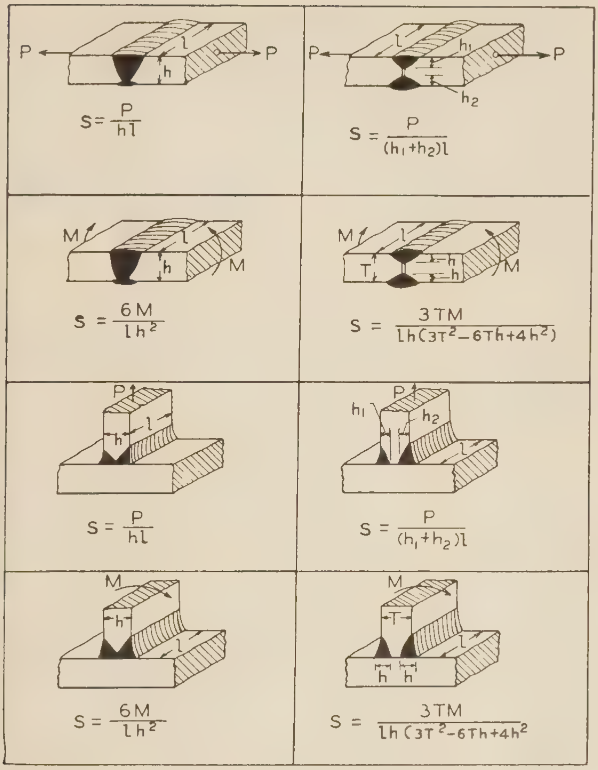


Fig. 19

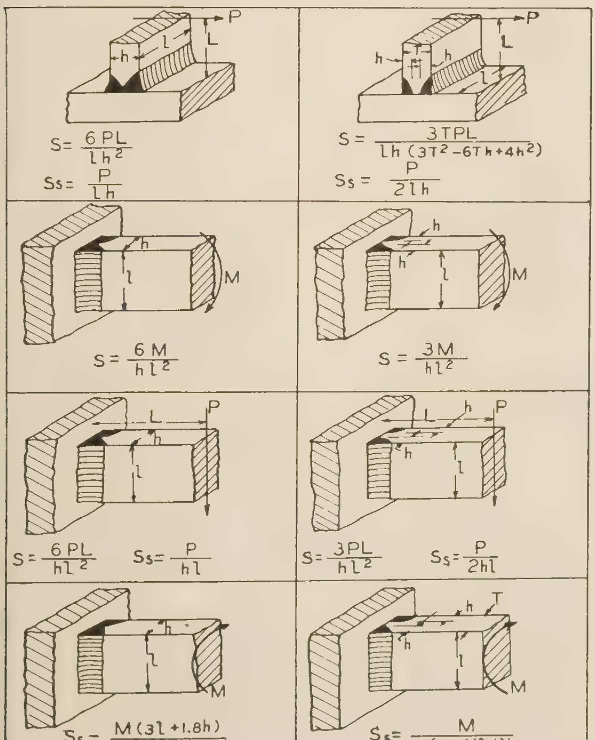


Fig. 20

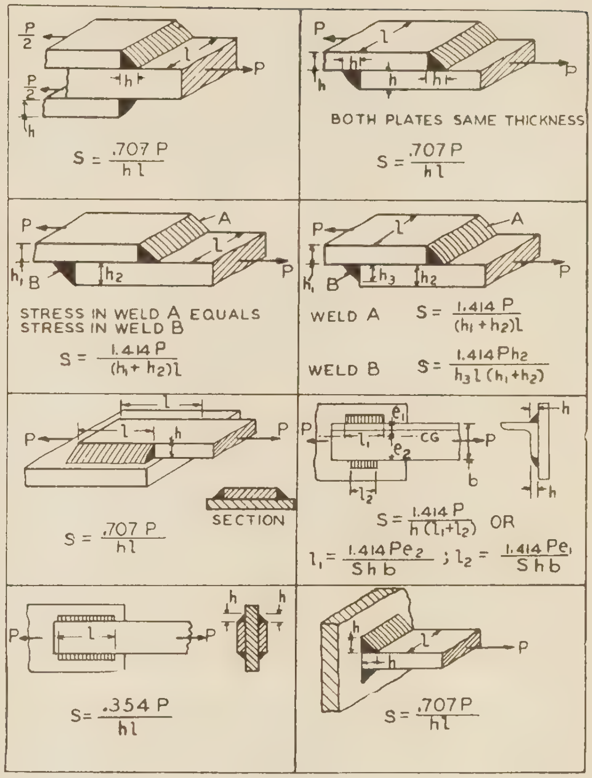


Fig. 21

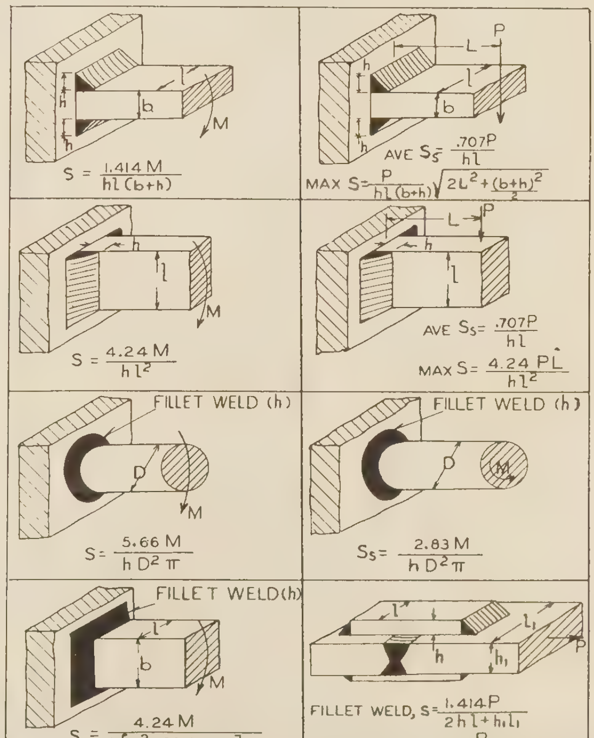


Fig. 22

large-diameter electrodes. On plates over 1 in. in thickness, it is advantageous to design the joint for small-diameter electrodes on the first passes.

When using fillet welds in any design it is important to remember that to double the size of a weld it is necessary to deposit four times as much weld metal. This fact often has an important influence in determining whether to use continuous or intermittent welding.

If a weld is designed on the basis of stress only, it is often possible that very small welds will be satisfactory. Experience has shown, however, that there is a minimum-size fillet weld that should be applied to a given plate thickness, if a sound strong weld is to be obtained. Recommended minimum-size fillet welds for different plate thicknesses, are given in Table 4.

When a continuous weld of the minimum size exceeds the required strength, and the weld is not required to be leakproof, intermittent welds may often be used. The minimum length of an intermittent weld should be at least four times the size of the fillet and never shorter than 1 in. A certain amount of time is required for a welder to start and stop a weld; consequently, it is recommended that welds longer than 1 in. be used whenever possible in order to reduce their cost.

Two types of intermittent welding are used, staggered and chain welding, as shown in Fig. 18. The choice between the two types is open to controversy. The staggered welding, however, has the advantage of producing a joint stiffness approximately equivalent to that of chain welding by using only half as much welding.

Recommended spacings for intermittent welds limit the maximum center-to-center spacing between increments to 16 times the thickness of the thinner member for compression, and 32 times the thickness of the thinner member for other loadings. In no case, however, should the spacing be greater than 12 in. between adjacent welds. (This spacing is somewhat greater than that permitted by the U. S. Navy on ship construction but it is entirely satisfactory for machinery and structures.)

In cases where two parts are lapped together and it is possible to use either parallel or transverse fillet welds, it is recommended that parallel welds be used because the load is generally more evenly distributed between the welds.

When fabricating such items as frames and bed plates from bars, plates, angles, or channels, it is generally preferred to use straight cut-off pieces rather than mitered ends to form the corners.

Bearing pads and other parts that require subsequent machining should have the welds designed strong enough to withstand the machining forces which may be larger than the service loads. Pads that have a width of over 12 times their thickness should be plug-welded at the center to prevent the center from bulging. The diameter of plug welds should be made from 2 to 4 times the thickness of the plate.

The general design of all structures should be such as to eliminate rigid and fixed joints as much as possible. Such joints tend to develop high internal stresses which will cause difficulty in fabrication and may impair the service life of the structure.

Equipment that requires close machining tolerances, and that will be subjected to dynamic service loads, should be stress-relieved after welding whenever possible. This stress-relieving process should consist of heating slowly to 1100 or 1200 F, soaking for 1 hour per inch of thickness and cooling slowly to at least 300 F before removing from the furnace.

Conclusion

It has been the aim of this paper to discuss briefly the essential factors of welding design. Many of the equations derived for computing the weld stresses are admitted to be approximate but experience has proved them to be adequate in all respects.

The working stresses and stress-concentration factors given are for welds made on mild low-carbon steels. When welding steels or materials of other types, other working stresses must be used. These values may be obtained from experimental tests in a similar manner.

Acknowledgment

The writer is indebted to R. E. Peterson, manager, mechanics division, Westinghouse Research Laboratories, for his interest and assistance in the preparation of this article.

Appendix

Figs. 19, 20, 21, and 22 on the preceeding page show a number of typical welded joints with the corresponding formulas for calculating the stresses. The formulas are based upon the methods of calculating stresses discussed in the paper.

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Welding Alloy Steels

By A. B. KINZEL,[1] NEW YORK, N. Y.

The welding of alloy steels is treated under three general headings; the low-alloy steels for general structural purposes, the heat-treated steels of the automotive type for machines and general engineering, and the high-alloy steels, such as stainless, for use in the chemical industries and the like. The effect of welding heat on metal adjacent to the weld proper and the intensity of internal stresses to be expected with joints of various design and various welding practices are discussed. It is shown that the butt weld has definite advantages from the engineering standpoint. The welding of heat-treated steels with austenitic welding rod and the welding of stainless steels with columbium-bearing welding rod are recent developments of note.

As engineering progresses, the use of specific materials best adapted to a given engineering purpose increases at a rate in keeping with the spirit of modern times. The ferrous materials having such specific properties are found almost exclusively in the category of alloy steels. Increased strength, corrosion resistance, wear resistance, and ease of fabrication are among the specific characteristics achieved to a greater degree by alloy steels. Not the least important problem in the use of these materials involves the ability to produce welded members or structures in which the welded joints have all the specific desirable features of the alloy steel proper.

Many of the recent types of alloy steels may be classified under the generic term, "low-alloy, high-strength." Steels to meet specifications involving 75,000 and 90,000 lb per sq in. minimum ultimate strength, together with ease of fabrication and good weldability, have been developed to a remarkable state of perfection during the past few years. Practically every steel company producing structural plate and sheet now offers a high-strength steel involving a low-alloy combination. The combinations of alloys in these steels differ widely but each of them successfully meets the requirement.

Low-alloy, high-strength steels as such are not new, having been used in bridges and special structural work for many years. The modern high-strength, low-alloy steels differ from their predecessors in that they may be welded without seriously affecting the metal next to the weld proper. This result is achieved through the use of chromium, manganese, or nickel, or some combination of these elements with silicon, vanadium, molybdenum, phosphorus, or copper, with carbon at a low level. From the fabricating standpoint it matters little which combination is used, providing that the alloy content is well-balanced and physical properties attained with carbon at 0.14 per cent or less in the 75,000 lb per sq in. grade and 0.22 per cent or less in the 90,000 lb per sq in. grades.

It is significant that almost without exception the current low-alloy, high-strength American steels approximating 75,000 lb per sq in. minimum tensile strength are obtained with carbon at 0.14 per cent or less, and the steels with a minimum ultimate strength of 90,000 lb per sq in. result from approximately 0.20 per cent carbon. Most of the low-alloy steel brands on the market today are made in both carbon ranges and follow this basic principle: The lower the carbon for a given strength, the higher the ductility and the lower the susceptibility to hardening under the heat-treating conditions imposed by various types of welding.

The welding of the steels in question presents problems common to them all. The first of these deals with the fluidity of the metal and the nature of the oxide and slag covering produced during the welding operation. The alloy content of each of the current low-alloy steels is sufficiently low so that no difficulty is evidenced on this score and the flowing properties and controllability of the weld puddle when using any of the better standard welding rods or electrodes are satisfactory for general-purpose welding. The problem which has been given the most serious consideration by the steel makers in their choice of analysis involves the air hardening of the steel in the zone adjacent to the weld. This air hardening may be definitely correlated with the carbon content and the ultimate strength of the steel. In general no difficulties are encountered due to the air hardening of steels containing 0.14 per cent or less carbon and having an ultimate strength less than 80,000 lb per sq in. as any increase in hardness due to the heat effect of welding is negligible. In the higher strength steels, with carbon at approximately 0.20 per cent, the air-hardening problem becomes serious with certain types of welding and must be given special attention.

It is significant that European development of low-alloy steels which has paralleled the American development in most respects differs from it radically in that a minimum ultimate strength of 73,000 lb per sq in. is specified and the 90,000 lb per sq in. grade is not currently manufactured and is not approved for use in general structural work in Europe. With the higher carbon grade the rate of cooling common to many welding operations is such that the allotropic transformation temperature of the steel is lowered to a point at which a hard structure is produced. This structure carries with it proportionate lowering of ductility so that in many cases a tempering treatment is requisite.

Residual internal stress is another problem involved in the welding of low-alloy steels which is definitely correlated with the air-hardening properties. This is of particular concern in the higher strength materials and a stress-relieving treatment which at the same time acts as a tempering treatment is frequently employed. Even the alloy steels in the 75,000 lb per sq in. class with carbon at 0.14 per cent or less which do not require

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tempering may, under certain welding conditions, require a stress-relieving treatment. It is well known that the zone involving temperatures from approximately 450 deg to the transformation temperature acts to relieve stress, combining as it does within its limits the standard stress-relieving temperature of 500 C. The upper limit of this zone is determined by the critical temperature of the steel on cooling and the width of the zone is determined by the temperature gradient involved in the welding operation. Moreover, the temperature of transformation is not only a function of the analysis of the steel but is further a function of the temperature gradient; the steeper this gradient, the lower the temperature of transformation, although the relation is not linear.

From the foregoing it will be seen that for a steel which possesses predetermined air-hardening properties, the magnitude of the stress-relieving zone on welding will be determined largely by the temperature gradient. With this in mind consideration of various types of welding and types of joints is in order.

From this point of view the gas-welded butt joint is the most satisfactory as the temperature gradient is less steep than with any other type of fusion welding currently in use. This is due to the fact that an appreciable portion of the plate metal is heated to the stress-relieving temperature zone so that the rate of cooling of any point next to the weld is definitely lowered. Electric arc-welded butt joints come next in the list. Here a small but definite zone next to the weld is heated above 500 C and this zone parallels the full thickness of the weld. The temperature gradient produced under these conditions, while steep, is of a much lower order of magnitude than that produced in any fillet joint. Fillet joints, due to their mechanical nature, carry temperature gradients far in excess of those obtained in the butt joints. Fillet joints, gas-welded, comprise but a small amount of metal at the edge of the fillet which has been heated above 500 C and electric-welded fillet joints comprise a still smaller cross-sectional area of metal at this temperature. In both instances the heat is transferred from a small area at the edge of the weld to the full thickness of the plate proper. The net result is to set up very steep gradients.

To summarize briefly, gas-welded butt joints are least prone to air hardening and are relatively free from internal stresses. Electric-welded butt joints are prone to air hardening when welding 90,000-lb steel and are subject to important internal stresses even when welding 75,000 lb per sq in. steel. The degree of this internal stress depends upon the mechanical design and location of the joint. All fillet welds tend to air harden to some degree. The amount of air hardening produced with gas or electric welding is small with a 75,000 lb per sq in. steel but with the 90,000-lb steel both gas and electric fillet welding produce air hardening to an appreciable degree. In gas-welded fillet joints the internal stress produced in 75,000 lb per sq in. steel is not serious but in electric-welded fillet joints in 75,000 lb per sq in. steel these stresses may be of sufficient importance to require stress relieving. Both gas- and electric-welded fillet joints in 90,000 lb per sq in. steel result in internal stresses making stress relieving mandatory in most instances.

Another factor involved in stress relieving which pertains particularly to low-alloy steels is comprised in the mechanism of stress relieving. Presumably stress relief takes place by plastic deformation of the metal and the rate of plastic deformation may be quantitatively indicated by short-time elastic limit or by creep values and the higher these values the less the deformation for a given stress. In the low-alloy high-tensile steels current today both of these values are higher than in carbon steels of equivalent tensile strength so that for a given temperature gradient a more severe internal-stress condition is to be expected.

Still another factor pertaining to internal stresses involves stress relieving by elastic deformation. This can only be effective if the flexural rigidity of the article or member is such as to allow local elastic deformation. This implies first that the member be constructed of thin material such as sheet metal and second that the form of the member be such as to allow the required motion without producing second-dimension stresses which exert undue restraint. A long, large-diameter sheet-metal cylinder is illustrative of the type of article in which stress relieving by stress distribution over a larger area due to elastic deformation is an important factor and a sphere of steel plate is illustrative of the type in which this plays a negligible rôle.

This discussion on internal stresses and air hardening is not intended to alarm the designing engineer but rather to point out that the problem does exist, that it does require consideration, and further, to point out those types of welding and types of steel to which most consideration must be given. The existence of thousands of feet of welding of all types in a wide variety of structures and applications is fitting testimony to the adequate consideration which fabricators have given to this problem. One other feature should also be stressed in this connection, namely, that the ductility of the base metal in the 75,000 lb per sq in. steels is so great that the most severe internal stresses are frequently relieved by room-temperature plastic deformation of the base metal proper. Thus these low-alloy 75,000 lb per sq in. steels have been aptly designated as fool-proof from the welding standpoint. The 90,000 lb per sq in. steels must, however, be used with discretion when welded structures are involved and whenever possible the stress-relieving tempering treatment should be applied to welded structures of these steels.

With the object of obtaining increased corrosion resistance, many of the low-alloy steels on the market today contain $^1/_2$ per cent or more of copper. This element not only aids corrosion resistance but also provides a moderate increase in tensile strength in the as-rolled condition. A further increase in tensile strength involving some 15,000 to 20,000 lb per sq in. may be obtained by heat-treating these steels in such a manner as to precipitate the copper from solid solution. However, the temperature range in which this precipitation is effective is comparatively narrow. The effect is completely destroyed by welding and if the copper is reprecipitated after welding the user continually faces the risk of losing the strength by local heating of the object in question after it has left the fabricator's hands. This local heating may occur accidentally or by intent in connection with repairs or minor alterations. As a result, conservative engineers and metallurgists have taken the position that the increased strength due to copper precipitation is not to be used in design.

While the low-alloy structural steels are of greatest interest at the moment, other alloy steels are also currently welded. Heat-treated S.A.E. steels may be joined by welding but in all cases a subsequent heat-treatment is necessary to insure optimum physical properties. The welding rod used to join these steels may be of the same analysis as the steel proper, but it is usual to use welding rods designed for optimum flowing properties, combined with satisfactory response to heat-treatment. It is quite common to weld chromium-nickel steel with chromium-molybdenum rod, or nickel-molybdenum steel with chromium-manganese-silicon rod. There seems to be very little advantage in using a rod of the same composition as the steel as this is only one way of fulfilling the single requirement that the response to heat-treatment of the deposited metal be of the same character as that of the metal being joined. The use of specially designed welding-rod analysis fulfills this requirement as well as the additional requirements imposed by the welding operation.

The welding of special steels in which corrosion and oxidation resistance rather than physical properties are the prime charac-

teristics presents a special case. Here it is necessary to have weld metal containing sufficient of the proper alloying ingredients to impart at least the same degree of chemical stability to the weld as is obtained in the base metal. However, even this requirement may be waived in special cases where the joints are so located as to receive preferential treatment with respect to chemical attack or in which the reinforcement of the weld metal is so heavy that a greater rate of chemical attack may be tolerated. For example, 4 to 6 per cent chromium steel may be welded with 4 to 6 per cent chromium-steel rod, but in many cases chromium-molybdenum rod, containing only 1 per cent chromium, has been found to be satisfactory.

Where heat-treatment following welding is not possible, the use of austenitic welding rod for joining ferritic steel is to be considered. This practice is now current in Europe. Welding rod containing 25 per cent chromium and 12 per cent nickel, or modifications of this analysis, which will result in austenitic deposited weld metal, gives such high ductility that internal stresses in the weld and adjacent base metal are reduced a very great degree. In addition this metal is supposed to protect adjacent metal from shock. In all such welding there will obviously exist a narrow zone of intermediate alloy content which will be martensitic in character and which will accordingly be definitely brittle. However, experience indicates that this zone is so narrow and is sufficiently discontinuous as to not affect the physical properties of the joint as a whole. Only further experience will tell whether this discontinuity together with the protection afforded by the weld metal proper can be adequately relied upon for engineering structures, but the experience to date is all in favor of this type of joint in those instances where the high cost of the welding rod can be tolerated.

The welding of austenitic stainless steel of the 18–8 type is another special case having wide ramifications. In such welding a specific phenomenon peculiar to austenitic steels is involved, namely, a precipitation in the temperature zone approximating 550 C immediately adjacent to the weld, which zone is rendered highly susceptible to chemical attack by this precipitation. A complete solution of the problem has been found in the addition of columbium to the low-carbon austenitic steels and in the use of columbium containing welding rod. Columbium in the welding rod renders the deposited metal free from the precipitation and attack in question and makes possible crossed welds in which every point is as free from corrosion as the base metal proper. This is due to the fact that the carbide-forming propensities of columbium prevent precipitation from solution in the temperature zone in question and the rate of oxidation of columbium is sufficiently low so that it can be adequately protected by a small amount of silicon in the welding rod proper. Thus the resulting deposited metal containing columbium uncombined with oxygen is clean, and free from intergranular corrosion.

From all this it will be seen that the welding of alloy steels resolves itself into a series of specific problems, that in each case these problems have been studied and solved, and that in the case of low-alloy steels, and austenitic stainless steels in particular, the solutions have been so successful that they may be considered to result in fool-proof welding steels and practices.

Arc Welding of Structural Alloy Steels

By W. L. Warner,[1] Watertown, Mass.

The author discusses the effect of welding heat on different structural alloy steels. This heat effect, expressed in terms of Vickers Brinell hardness, is affected by the physical and chemical properties of both the parent metal and the electrode used. The author presents graphs showing the extent of the effect of these properties on the welded structure and on the weldability of the material.

The ease or the difficulty with which a steel structure may be fusion-welded successfully depends upon several factors including: (1) The chemical composition of the steel, and (2) the heat effect of welding on the steel. There are, of course, other factors involved, such as heat-treatment of the steel, design of the structure, the type of service for which the structure is intended, and the availability of suitable welding materials and personnel, but the two first mentioned are of fundamental metallurgical importance.

The application of the fusion-welding method in any form necessarily involves the use of heat in order to produce fusion. This requirement is unavoidable. This heat of welding affects certain portions of the steel in the immediate vicinity of the weld in the process of depositing the molten metal which forms the weld. The zone of the base metal which is affected by the heat of welding is called the "heat-affected zone."

When considering this zone adjacent to the weld which has been affected by the heat of welding, it must be remembered that the width of the heat-affected zone and the degree of hardness formed are dependent upon the mass of the piece, the welding energy used, and the amount of heat input per unit of length of weld or the rate of travel of the welding arc, as well as the plate composition. Therefore, in making comparisons between different steels as to heat effect of welding, such factors as the mass of the pieces and the welding conditions must be similar or incorrect comparisons will be obtained. Also, it is desirable to make the comparisons of heat effect under conditions which give as nearly as possible the steepest temperature gradient obtainable under practical welding conditions.

The practice at the Watertown arsenal is to make comparisons of heat effect on plate metal $^1/_2$ in. thick. Single beads of weld metal are laid approximately 3 in. long on a piece of plate approximately 3 in. wide and 9 in. long. The bead is laid lengthwise of the piece approximately in the center, and when cold, a piece about $^1/_2$ in. wide is cut through the bead and plate about 1 in. from the start of the bead. This piece is surface-ground on both cut surfaces to give true parallel faces for hardness measurements by the Vickers method. Figs. 1, 2, and 3 show Vickers hardness surveys taken on sections of weld beads run with both slow and fast welding speeds.

Comparisons of heat effect on different steels are generally made in the as-welded condition, and subsequently the effect of stress relieving or other heat-treatment is determined, but the basic comparison is made on the as-welded specimen. There are good and valid reasons for this procedure.

Determinations of heat effect on cross sections of multiple-layer welds give valuable and interesting data, but the result obtained shows the final heat effect due to the multiple-layer procedure. A heat-effect test of this nature does not usually show the maximum hardening effect on the base metal, which occurs when the first layer is deposited on the cold plate, and it is at this point in the making of a weld that cracks may start due to the heat effect on the base metal. The heat of subsequent layers, if applied within a reasonable time, will greatly reduce the heat effect of the first layer applied. Hence, the result obtained from a hardness survey on the cross section of the finished weld does not indicate how dangerous or how critical may have been the situation when the first layer was applied.

Therefore, in this study of heat effect on the base metal, we are trying to determine the maximum hardness formed in the heat-affected zone under the worst conditions of rapid heating and cooling, for after the weld has been finished and given a stress-relieving heat-treatment, the danger of plate or weld cracking due to high hardness in the heat-affected zone is past.

The chemical composition of the base metal also has a pronounced affect on the hardness formed in the heat-affected zone. The element mostly responsible for the hardening is carbon. The metallic elements are contributory only and serve rather to intensify the hardening effect of the carbon than as hardening elements themselves.

In Fig. 4 are shown hardness curves of weld metal and of the heat-affected zone on plain carbon steels ranging from about 0.15 to 0.50 per cent carbon. These weld beads were deposited on flat plate $^1/_2$ in. thick, 9 in. long, and approximately 3 in. wide. The weld beads were laid the entire length of the 9-in. piece by automatic welding, using a high current of the order of 350–400 amp. Therefore, the results shown are not directly comparable with those shown in Figs. 1, 2, and 3, which are hand welds, because the amount of heat put into each specimen by the automatic welds was far greater than that put into the hand-welded specimens; although the plate pieces used in both cases were the same size. If, instead of running the weld beads the entire length of the 9-in. piece, the beads had been run only 3 or 4 in. long in the center, it is the author's opinion that the curves shown in Fig. 4 would have shown considerably higher hardnesses even in the lower carbon range.

The curves shown in Fig. 4 have a considerably steeper slope in the higher carbon range from 0.30 to 0.50 per cent than in the range from 0.15 to 0.30 per cent. The author believes that this difference is similarly true of nearly all steels as far as heat effect of welding is concerned. Any one who believes that fifteen points of carbon do not have much effect on the hardness of the heat-affected zone, therefore, must qualify his statement before

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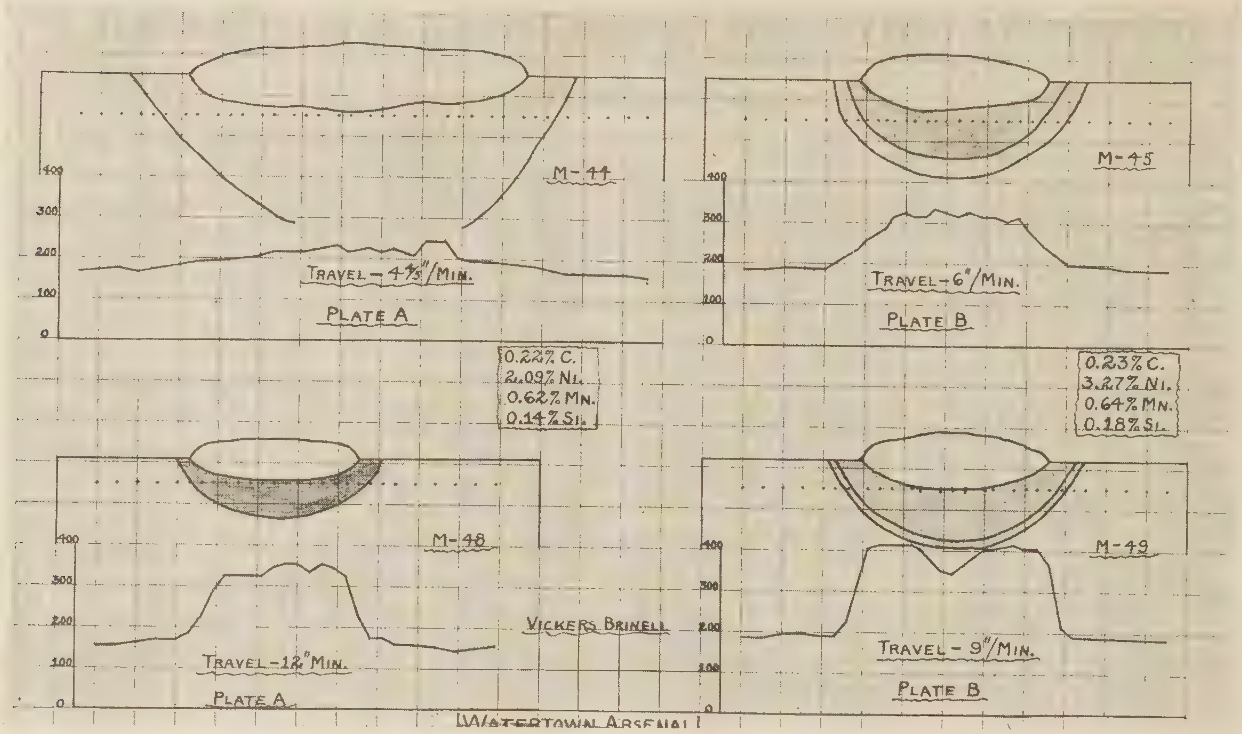


Fig. 1 Effect of Welding Heat on 1/2-In. Ni Steel Plate

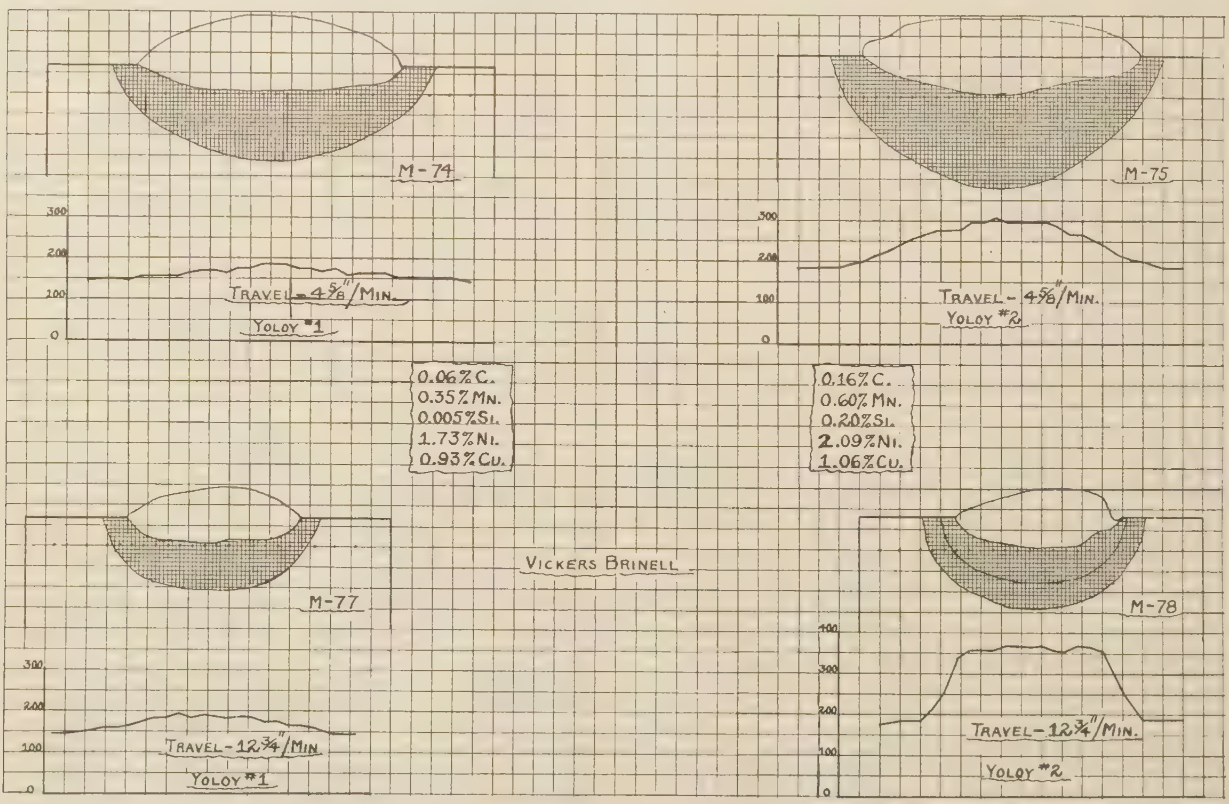


Fig. 2 Effect of Welding Heat on 1/2-In. Ni-Cu Steel Plate

TABLE 1 CHEMICAL AND PHYSICAL PROPERTIES OF PLATE AS ROLLED[a]

Plate		Chemical composition								Physical properties, not welded							
										Tensile strength, lb per sq in.		Elastic limit, lb per sq in.		Elong. in 1 in., %		Tensile impact, ft-lb	
No.	Material	C	Mn	Si	Ni	Mo	Cu	P	Cr	As rolled	Stress rel'd	As rolled	Stress rel'd	As rolled	Stress rel'd	As rolled	Stress rel'd
1	3½% Ni	0.35	0.62	0.16	3.37	..	..	..	0.22	107,000	100,000	55,000	61,000	35.0	40.0	810.0	820.0
2	3½% Ni	0.23	0.64	0.18	3.27	..	..	..	..	90,000	84,000	50,000	57,000	40.0	50.0	750.0	768.5
3	2% Ni	0.22	0.62	0.14	2.09	..	..	..	..	80,000	75,000	42,000	50,000	50.0	60.0	790.0	829.7
4	Cu-Ni-Mo	0.23	0.72	0.02	0.79	0.15	1.56	0.02	..	105,000	89,000	58,200	60,000	40.0	48.0	773.0	778.3
5	Cu-Ni-Mo	0.09	0.85	0.03	1.08	0.10	1.60	0.02	..	86,650	73,000	40,800	50,000	46.5	55.0	740.0	706.1
6	Cu-Ni	0.06	0.35	0.005	1.73	..	0.93	..	..	70,700	67,700	48,400	45,200	56.0	61.0	768.5	730.1
7	Cu-Ni	0.16	0.60	0.20	2.09	..	1.06	..	..	87,300	80,500	54,000	56,000	50.0	51.5	811.8	797.2
8	Cr-Cu-P	0.12	0.40	0.76	..	..	0.39	0.16	1.08	75,800	75,000	43,800	46,200	59.0	59.0	875.3	836.2[b]
9	Mn-Cu	0.30	1.40	0.20	..	..	0.25	..	..	91,400	80,300	45,400	40,000	53.0	58.0	934.0	860.7
10	Mn-Si	0.36	0.76	0.26	..	..	0.03	..	..	96,400	88,500	35,800	36,200	42.5	45.5	894.8	875.3
11	Mn-Mo, no. 1	0.17	1.57	0.43	..	0.38	..	0.02	..	104,000	92,000	36,000	47,000	42.0	46.0	890.0	800.0
12	Mn-Mo, no. 2	0.12	1.00	0.21	..	0.49	..	0.03	..	84,000	80,000	37,000	36,500	54.0	51.0	780.0	800.0
13	Cu-Mo	0.29	0.74	0.24	..	0.22	0.35	0.03	..	90,000	88,000	45,000	50,000	40.0	42.0	810.0	820.0

[a] Each value given in the table is the average of two test specimens.
[b] This value obtained from one specimen only.

TABLE 2 PHYSICAL PROPERTIES OF ARC BUTT WELD AND CHEMICAL PROPERTIES OF ALLOY ELECTRODES[a]

	Butt welds[b]								Electrodes			
	Tensile strength, lb per sq in.		Elastic limit, lb per sq in.		Elong. in 1 in., %		Tensile impact, ft-lb		Alloy, %		Carbon in core wire,	
Plate no.	As welded	Stress rel'd	As welded	Stress rel'd	As welded	Stress rel'd	As welded	Stress rel'd	Ni	Mo	per cent	Coating
1	104,000	....	55,000	....	20.5	..	560.0	...	2.5	0.3	0.15	Mineral
2	95,500	....	55,000	....	17.0	..	600.0	...	2.5	0.3	0.15	Mineral
3	85,000	....	49,500	....	23.0	..	670.0	....	2.5	0.3	0.15	Mineral
4	99,250	....	53,000	....	22.0	..	581.6	...	...	0.5	0.13	Organic and mineral
5	85,500	....	44,400	....	32.5	..	688.4	...	...	0.5	0.13	Organic and mineral
6	.....	....	....	....	..	..	...	...	...	...	..	....
7	.....	....	....	....	..	..	...	...	...	...	..	....
8	85,350	78,000	46,600	41,600	25.0	26.0	744.4	707.0	...	0.5	0.13	Organic and mineral
9	97,100	87,000	50,000	35,400	26.5	36.0	725.6	641.5	...	0.5	0.13	Organic and mineral
10	97,400	92,000	48,000	44,000	22.5	36.0	698.1	744.6	...	0.5	0.13	Organic and mineral
11	94,870	93,300	55,800	....	22.0	36.5	614.0	704.0	...	0.5	0.13	Organic and mineral
12	85,600	83,800	51,600	50,000	29.5	38.0	712.2	787.7	...	0.5	0.13	Organic and mineral
13	93,750	86,500	51,200	46,800	37.0	39.0	697.1	753.7	...	0.5	0.13	Organic and mineral

[a] Each value given is the average of two specimens.
[b] Tests not completed.

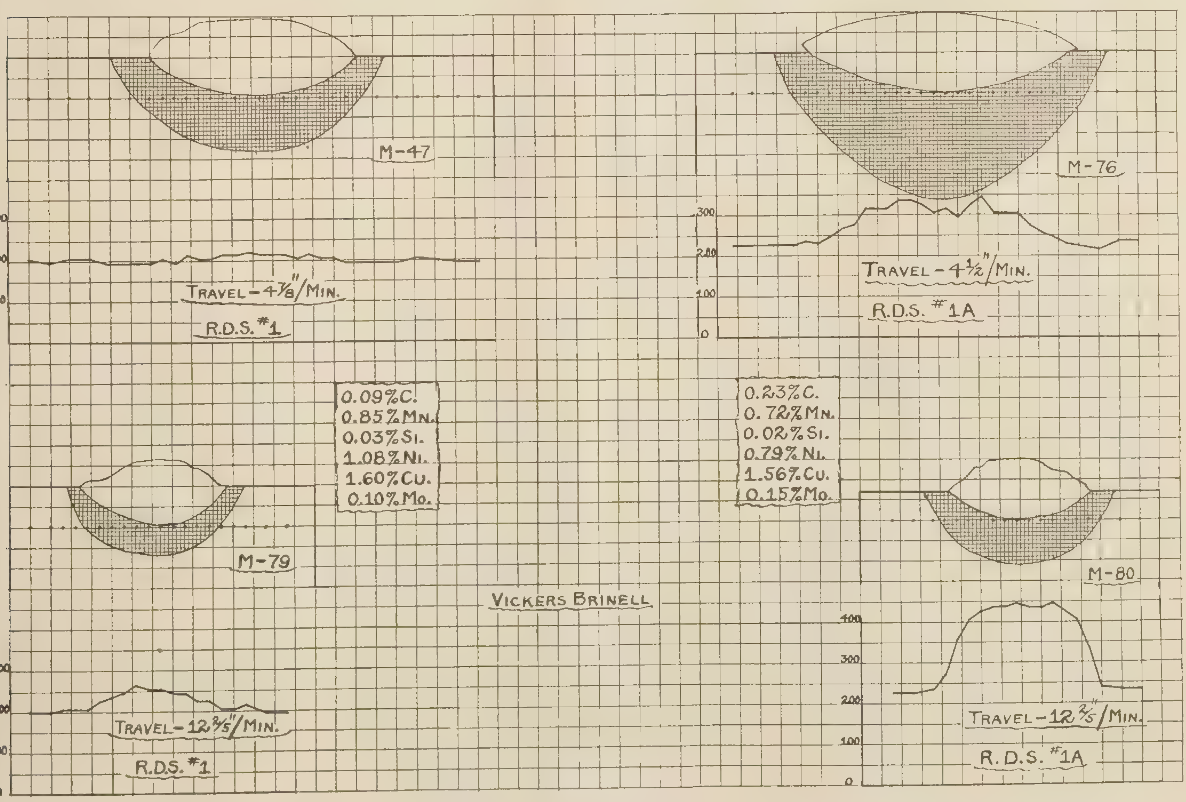


FIG. 3 EFFECT OF WELDING HEAT ON ½-IN. NI-CU-MO STEEL PLATE

expressing himself publicly to that effect. A stress-relieving heat-treatment at 600 C will reduce the hardness of the heat-affected zone to a level slightly higher than the plate metal as rolled.

At the Watertown arsenal the welding characteristics of several of the new low-alloy steels are being studied and compared with structural nickel steel as used by the Ordnance Department of the U. S. Army for the building of gun carriages. The specified physical requirements for this material have been published elsewhere.[2] The method of testing used to determine the tensile properties of the welded joint is as described in that article,[2] although some changes have been made in the shape of the tensile impact specimen for butt joints. The dimensions of the test

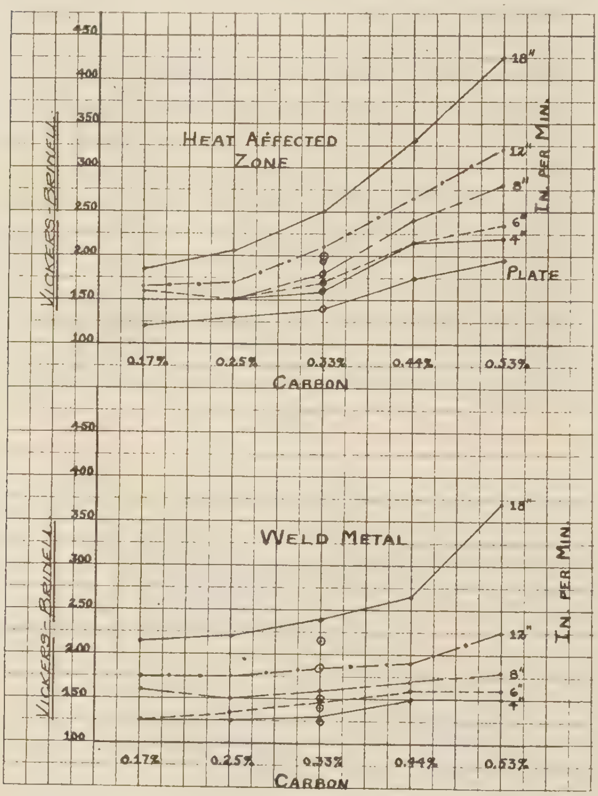


Fig. 4 Effect of Heat From Automatic Welding on $^1/_2$-In. Plain Carbon-Steel Plates
(Courtesy of Dr. W. G. Theisinger and Harvard University.)

specimen now used for butt joints are shown in Fig. 5. These dimensions hold for a single V butt joint having a 30-deg bevel or a single U butt joint. The general rule for determining the length of the straight test section is to add $^1/_4$ in. to the width of the weld, or to add $^1/_2$ in. to the width of the groove.

In Table 1 are shown the chemical composition and tensile properties of unwelded $^1/_2$-in. structural alloy-steel plate metals which have been tested at the Watertown arsenal. Table 2 shows the tensile properties obtained from arc butt-welded joints on these plates.

Three commercial types of electrodes, the properties of which are given in Table 3, were used for these welding tests. The properties of the electrode used in welding the structural nickel steels are also given in Table 3.

[2] "Welding of Structural Nickel Steel," by W. L. Warner, *The American Welding Society Journal*, vol. 13, June, 1934, pp. 15–23.

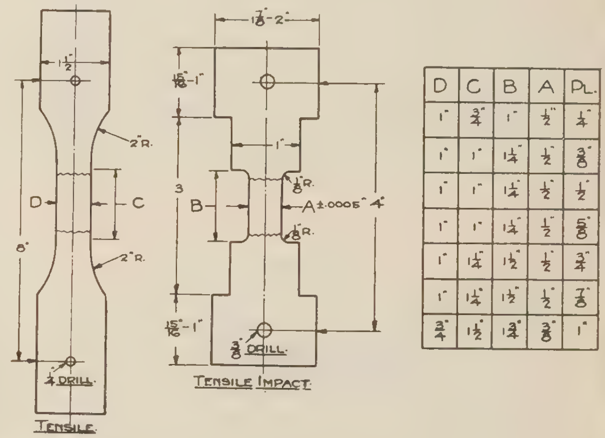
TABLE 3 PROPERTIES OF ELECTRODES USED IN WELDING THE ALLOY-STEEL PLATES

Plate	Electrode type	Carbon in the core wire, per cent	Coating	Alloy in the weld deposit, per cent Mo	Ni
Low-alloy steel	1	0.15	Organic and mineral	0.5	. . .
	2	0.25	Mineral	0.5	. . .
	3	0.15	Mineral	. . .	. . .
Ni steel	. .	0.15	Mineral	0.3	2.5

TABLE 4 VALUES OF MAXIMUM HEAT EFFECT

Plate[a]	Carbon, per cent	Vickers Brinell hardness after welding with arc speeds of— 4 to 6 in. per min	10 to 12 in. per min
1	0.35	490	540
2	0.23	330	410
3	0.22	240	360
4	0.23	340	450
5	0.09	220	260
6	0.06	190	190
7	0.16	310	370
8	0.12	270	. . .
9	0.30	350	. . .
10	0.36	320	. . .
11	0.17	287	397
12	0.12	221	270
13	0.29	258	413

[a] See Table 1.



D	C	B	A	PL.
1"	3/4"	1"	1/2"	1/4"
1"	1"	1 1/4"	1/2"	3/8"
1"	1"	1 1/4"	1/2"	1/2"
1"	1"	1 1/4"	1/2"	5/8"
1"	1 1/4"	1 1/2"	1/2"	3/4"
1"	1 1/4"	1 1/2"	1/2"	7/8"
3/4"	1 1/2"	1 3/4"	3/8"	1"

Fig. 5 Dimensions of Tensile-Test Specimen for Butt Welds as Used at the Watertown Arsenal

All of the electrodes listed in Table 3 gave good arcing characteristics and sound welds with the exception of electrode No. 3. On the copper-nickel steels and those containing high phosphorous, chromium, and copper, electrode No. 3 gave a porous weld metal. However, the tensile properties of this porous weld metal were good, in some cases nearly equaling those obtained with the alloy electrodes. Figs. 6 and 7 show data on two grades of steel from which a comparison between the low-carbon and the alloy electrodes may be made. The low-carbon weld values are shown by the two blocks at the right in each group of each figure as indicated thereon. On the other steels tested, the comparison between electrodes was quite similar.

In Table 4 are shown the maximum hardnesses set up in the heat-affected zones of the various plate materials listed in Table 1. These hardness values have been determined by the procedure previously described and illustrated in Figs. 1, 2, and 3.

If these maximum hardnesses are compared with the corresponding plate composition as given in Table 1, some conception is obtained of the hardening effect of the alloying elements and the relation between this hardening effect and the amount of the element present in the steel. Such a comparison is shown in Fig. 8. At the top of the left-hand column are shown the values of maximum hardness found in the heat-affected zone with the

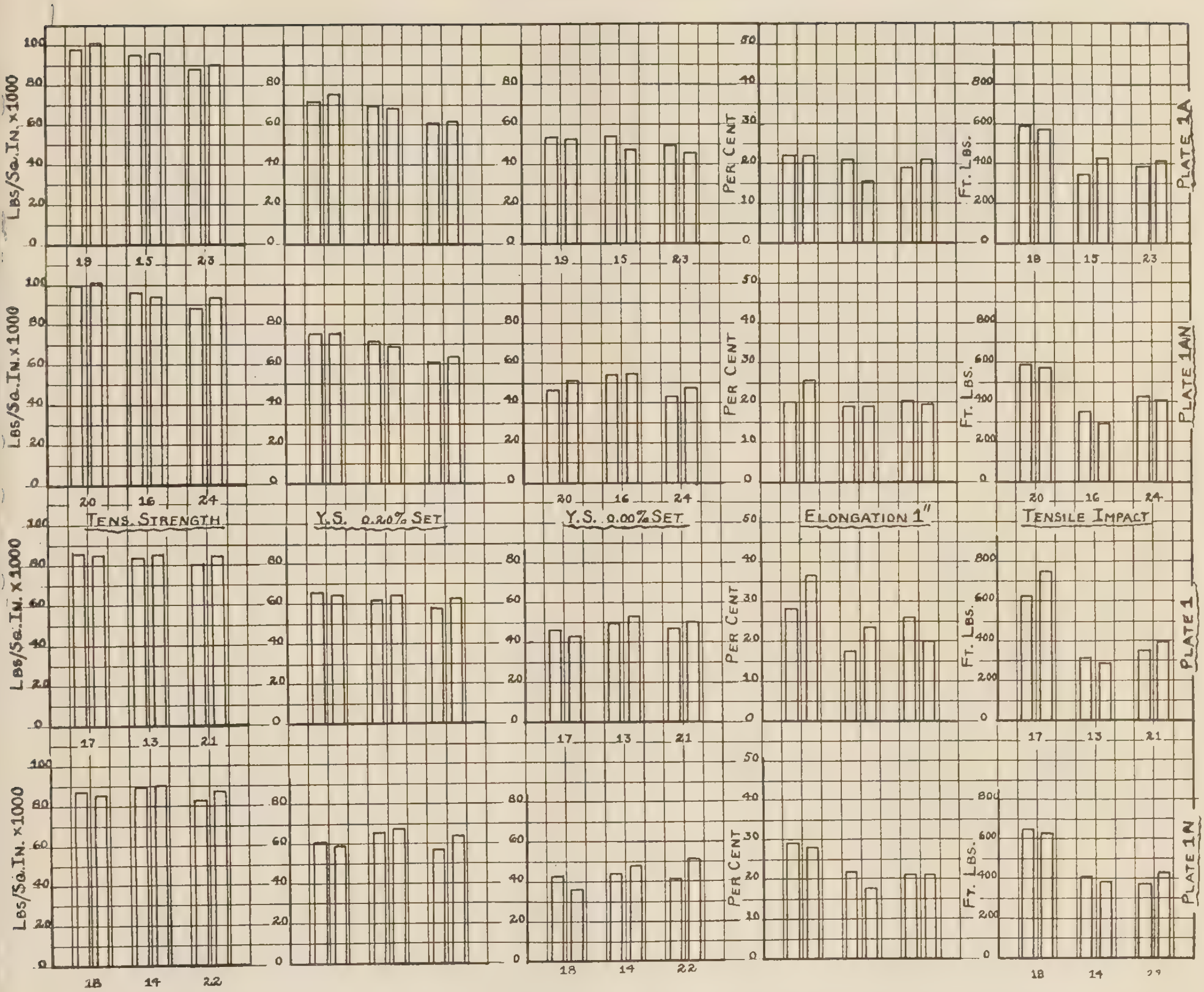


Fig. 6 Physical Properties of Butt Welds Made With Three Types of Electrodes on Four Different $^{1}/_{2}$-In. Cu-Ni-Mo Steel Plates

(Numbers 13, 14, 15, and 16 indicate Toncan metal electrode. Numbers 17, 18, 19, and 20 indicate 0.50-Mo steel electrode. Numbers 21, 22, 23, and 24 indicate low-carbon steel electrode.)

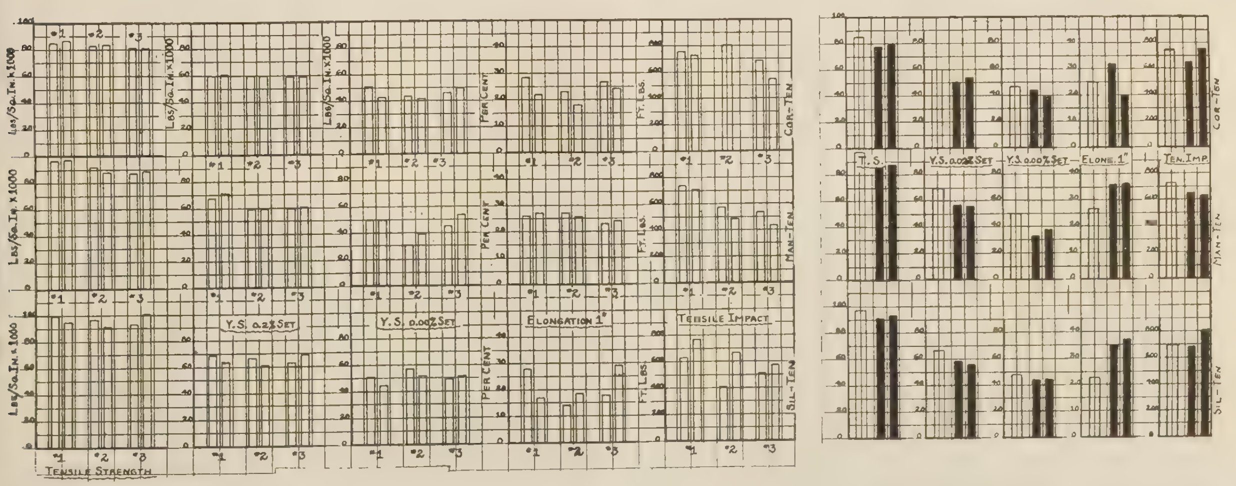


Fig. 7 Physical Properties of Butt Welds Made With Three Types of Electrodes on $^{1}/_{2}$-In. Alloy-Steel Plate

(Numbers 1, 2, and 3 indicate 0.50-Mo, 0.50-Mo, and low-carbon electrodes, respectively. Results shown by the right-hand group of graphs were obtained by using a 0.50-Mo electrode.)

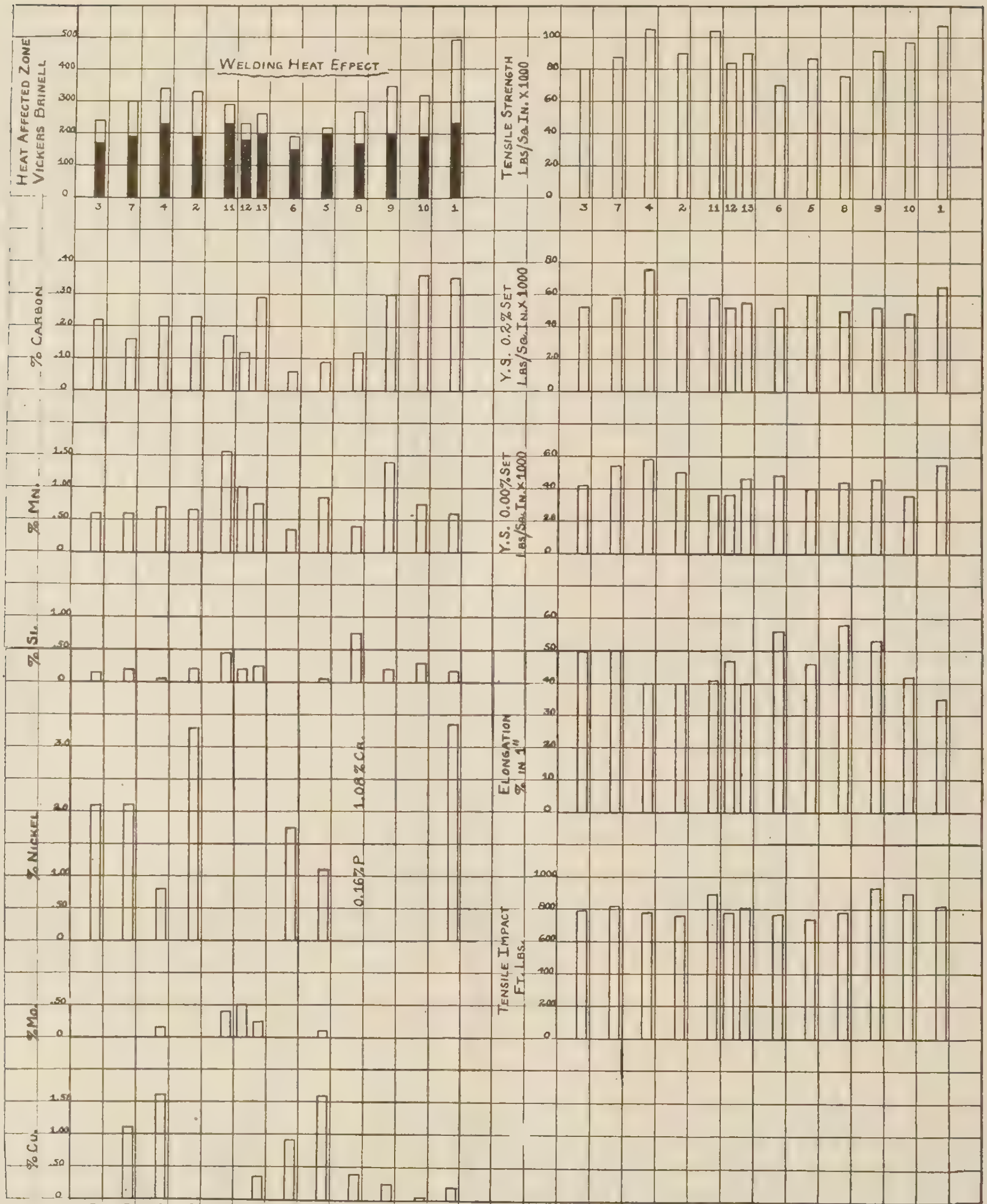


Fig. 8 Left: Relation of Chemical Composition of 1/2-In. Alloy Plate As Rolled to the Effect of Welding Heat. Right: Physical Properties of the Plate As Rolled

slow arc-travel speed of 4 to 6 in. per min, which is the approximate normal speed for hand welding. The shaded portion of each block represents the plate hardness unaffected by the heat of welding. The remainder of the left-hand column shows the amounts of the various alloying elements in the plate. The identification of the plates is shown at the bottom of the left-hand column. Physical properties of the plate as rolled are shown in the right-hand column.

It will be noted that the heat effect of welding varies nearly as much as the carbon content, and also that this variation is in no respect similar to the variation in content of any of the other elements shown. This point is believed significant as these data indicate that the heat effect of welding obtainable, as shown by the maximum hardness found, is determined mainly by the carbon content of the plate metal. The presence of metallic elements may affect the hardness value to some extent, but the effect of the carbon is of most importance. If a so-called weldable alloy steel is desired, the carbon content should be limited to a safe value which the author suggests as 0.25 per cent.

It is, of course, possible to weld a high-carbon steel or high-alloy steel by using special precautions and procedure, but by the term "weldable," the author understands that such a steel is to be capable of practical welding, easily and without the necessity for preheating or other heating or cooling precautions to avoid cracking and excessive hardness in the heat-affected zone.

A summary of heat-effect data on structural nickel-steel plate is shown in Fig. 9. The carbon content varies from approximately 0.20 to 0.45 per cent with a 2 and a $3^1/_2$ per cent nickel

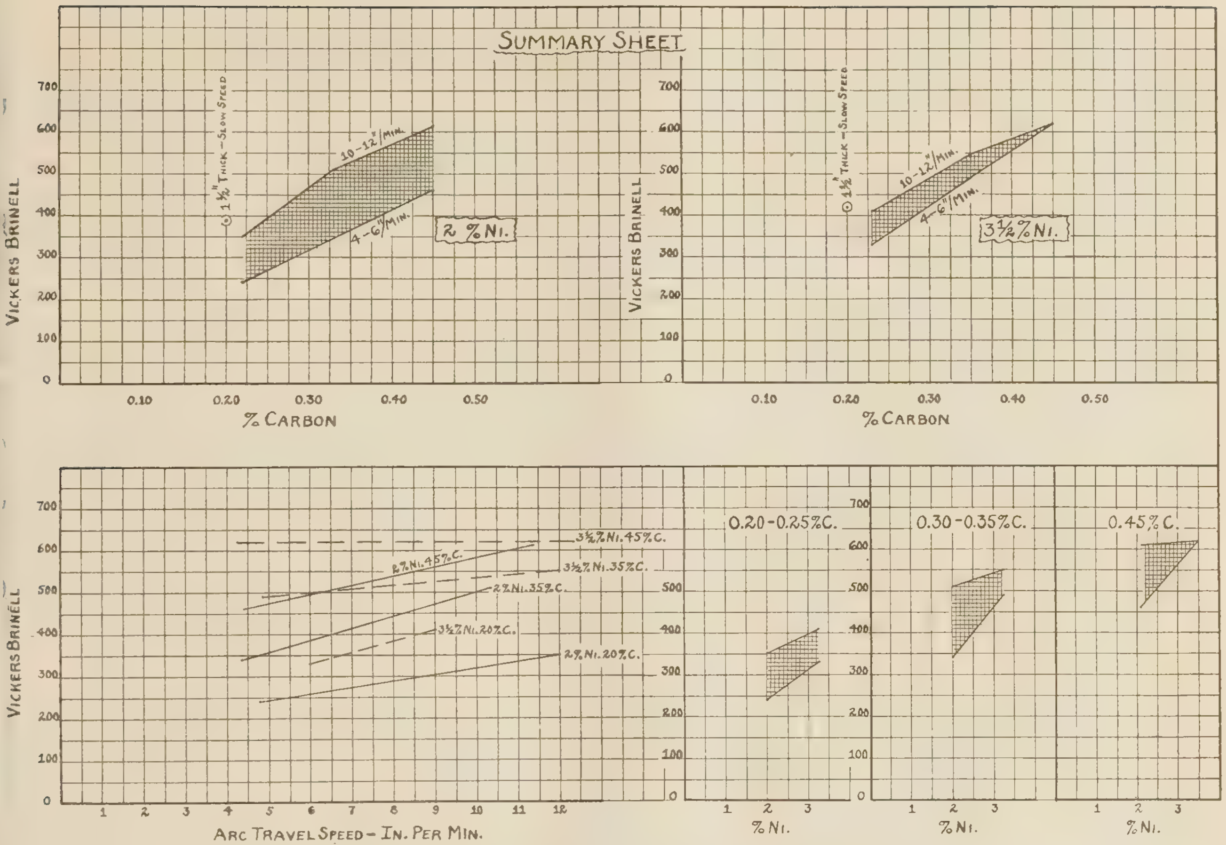


FIG. 9 EFFECT OF WELDING HEAT ON 1/2-IN. STRUCTURAL NICKEL-STEEL PLATE
(Courtesy of the International Nickel Company.)

content, respectively. These data show the pronounced effect of the carbon on the hardness of the heat-affected zone.

Nickel is sometimes considered as a hardening element in steel and should therefore be very limited in amount for weldability. A study of the data in Fig. 9 will show that when the carbon content is less than 0.25 per cent, the hardness in the heat-affected zone is not seriously affected by the nickel content. Also, if the averages of the shaded areas in the upper half of Fig. 9 were considered, it would be found that for 2 and $3^1/_2$ per cent nickel content, the average hardness would be 430 and 500, respectively. Thus, for a 75 per cent increase of nickel, the average hardness increases about 15 per cent. If now we consider the 2 per cent nickel alone and compare the average hardness for the minimum and maximum carbon content, it is seen that for carbon contents of 0.22 and 0.45 per cent, the average hardness is 300 and 540, respectively. Thus, for a 100 per cent increase in carbon content there is an 80 per cent increase in the hardness of the heat-affected zone when 2 per cent of nickel is present. The same comparison when $3^1/_2$ per cent nickel is present shows that with 0.22 and 0.45 per cent carbon, the average hardness is 370 and 600, respectively, or an increase of 65 per cent in the hardness of the heat-affected zone with a carbon increase of 100 per cent. These figures clearly indicate that the carbon content, rather than nickel, is of primary importance when considering the weldability of structural nickel steel.

It is regretted that the author has not sufficient data on steels containing other metallic elements as a major item along with carbon so that a similar comparison can be made. Data of this sort are of great importance not only to the designer of welded structures and the welding engineer who is concerned with the building of them, but also to the steel maker who must make steel plate suited to the requirements of the users of welded structures.

If we may assume, for the sake of argument, that the maximum hardness of the heat-affected zone is an index of the weldability of the steel plate, i.e., the lower the hardness in the heat-affected zone, the better is the weldability of the steel, and plot the tensile properties of the plate against this hardness, then we may obtain some conception of the desirability of various plate compositions for different design requirements.

In Fig. 10 are shown the values given in Table 1 plotted against the maximum hardness in the heat-affected zones as given in Table 4. The graphs in each of the figures have been arranged in five groups according to carbon content. If we consider the tensile strength and proportional (elastic) limit values shown, it is apparent that with a carbon content of 0.30 per cent or over, an increase of strength cannot be obtained without a considerable increase in the hardness of the heat-affected zone or, in other words, decreased weldability.

With a carbon content up to 0.22 per cent, the average slope of the curves is less than for carbon contents of 0.30 and 0.35 per cent, so that an increase of tensile strength or proportional (elastic) limit can be obtained with the lower carbon alloys at a much smaller sacrifice of weldability. This appears to the author as a fundamental metallurgical principle applicable to the welding qualities of structural alloy steels, the only difficulty being that in this study there are too few compositions in any one group.

If, therefore, this welding principle is true, then increased

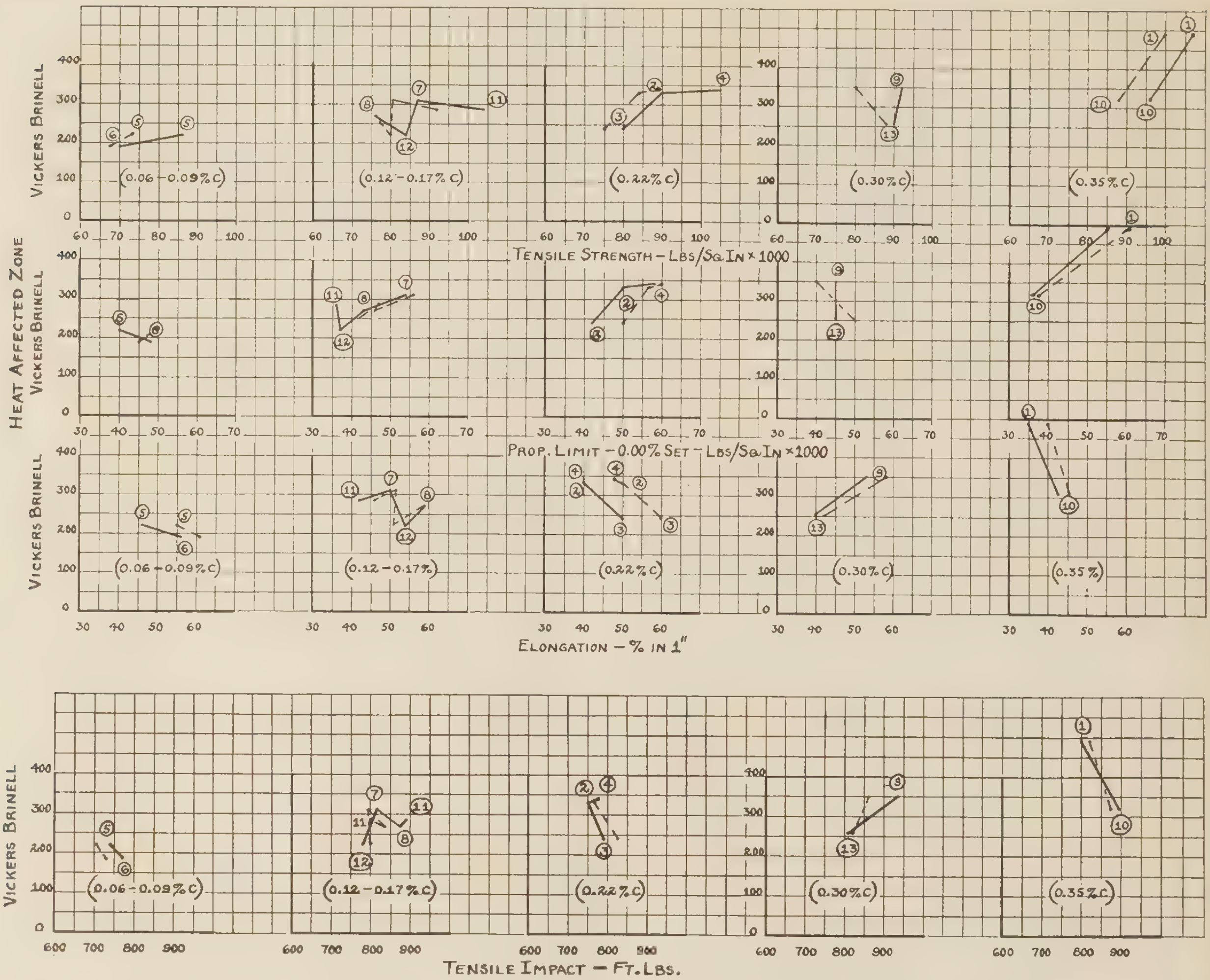


Fig. 10 Effect of Welding Heat Vs. the Physical Properties of 1/2-In. Low-Alloy-Steel Plate
(Numbers in circles refer to Table 1. Solid line indicates results from steels as rolled. The broken lines indicate results after stress relieving at 600 C.)

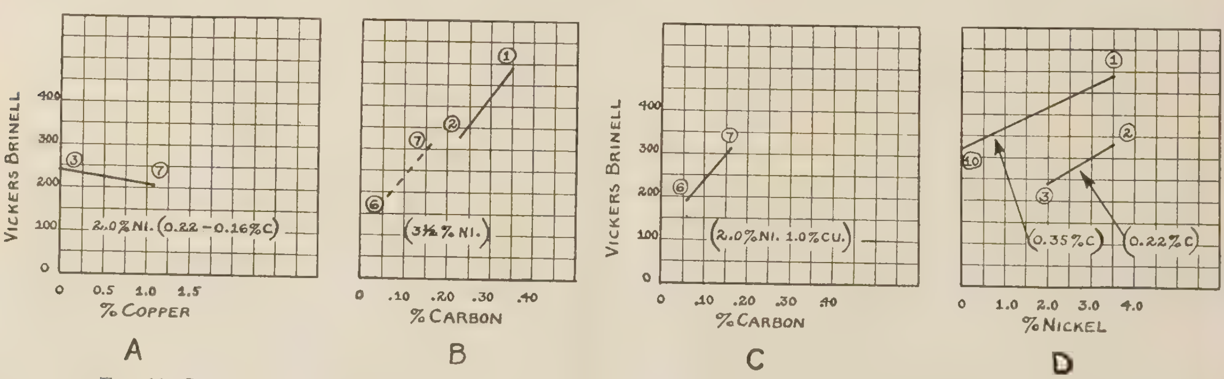


Fig. 11 Influence of Chemical Composition of 1/2-In. Low-Alloy-Steel Plate on Welding-Heat Effect

tensile properties should be obtained, not by increasing the carbon content, but rather by increasing the alloy content of the steel and at all times the carbon should not be over 0.25 per cent for good weldability.

With reference to Fig. 10, the values for elongation and tensile impact appear to increase as the value for heat effect decreases, or weldability increases. This is as it should be. However, there is one exception to this rule, steel No. 9, which is in the range of 0.30 per cent carbon, shows an increase of these two properties with an increase of hardness or decrease of weldability. This particular steel is of high manganese content and no definite reason can be suggested for this apparent peculiarity.

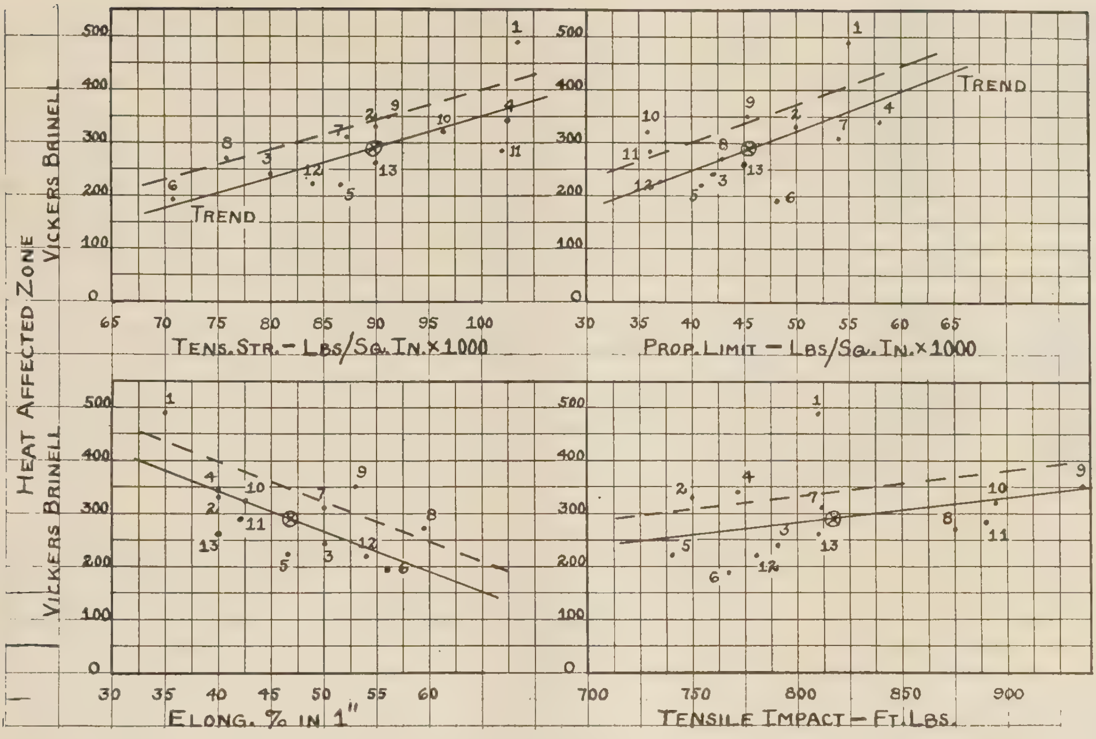


FIG. 12 EFFECT OF WELDING HEAT VS. THE PHYSICAL PROPERTIES OF 1/2-IN. LOW-ALLOY-STEEL PLATE AS ROLLED

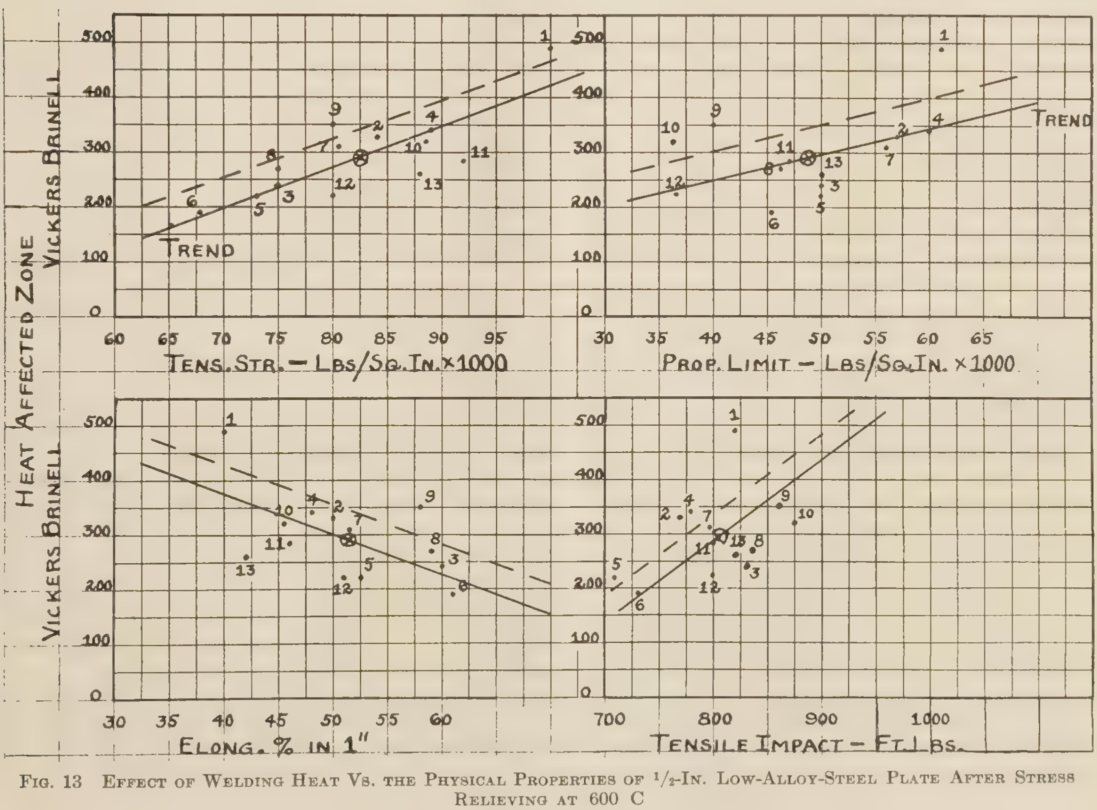


FIG. 13 EFFECT OF WELDING HEAT VS. THE PHYSICAL PROPERTIES OF 1/2-IN. LOW-ALLOY-STEEL PLATE AFTER STRESS RELIEVING AT 600 C

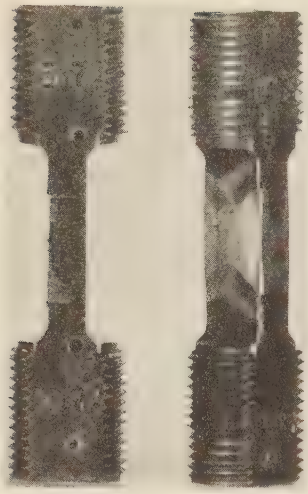


Fig. 14 Low-Temperature Tensile Impact-Test Specimen for Butt-Welds

In Fig. 11 are shown a few apparent relations between alloy content and heat effect of welding. The effect of copper indicates a decrease in hardness with an increase of copper in the presence of 2 per cent nickel. In this particular case, however, the carbon content also dropped from 0.22 to 0.16 per cent, so that the curve does not show the effect of the copper alone.

The effect of the carbon is shown in Fig. 11*B* and *C*. Note that the slope of the curves is slightly greater than 45 deg, which indicates a rapid increase of hardness with increasing carbon. Note here that the slopes of the two curves are practically identical, indicating that the effect of the carbon on the weldability is independent of the metallics present in the steel in this case.

Further, if we transpose the curve in Fig. 11*C* to the coordinates of Fig. 11*B*, it is seen that probably the steel containing 1.0 per cent copper, were the carbon to be increased to about 0.30 per cent, would have a slightly greater hardness than the $3^1/_2$ per cent nickel steel. In this comparison about $1^1/_2$ per cent of nickel is replaced by 1.0 per cent of copper with a resultant apparent greater hardness. However, it is possible that alloying elements in combination with each other have quite different effects than when acting alone in the steel. In any case, the effect of the carbon is self-evident.

By plotting the tensile properties of these steels together on one set of coordinates as shown in Figs. 12 and 13, the steels may be more easily compared. The average values, represented by the solid line, indicate a gradual increase of strength as the hardness of the heat-affected zone increases. If a maximum hardness of 50 points in excess of the average is allowed, then the broken line indicates an arbitrarily selected upper limit of acceptability for welding purposes.

Considering this method of testing only, these materials might be listed in the order of desirability shown by Table 5. It should

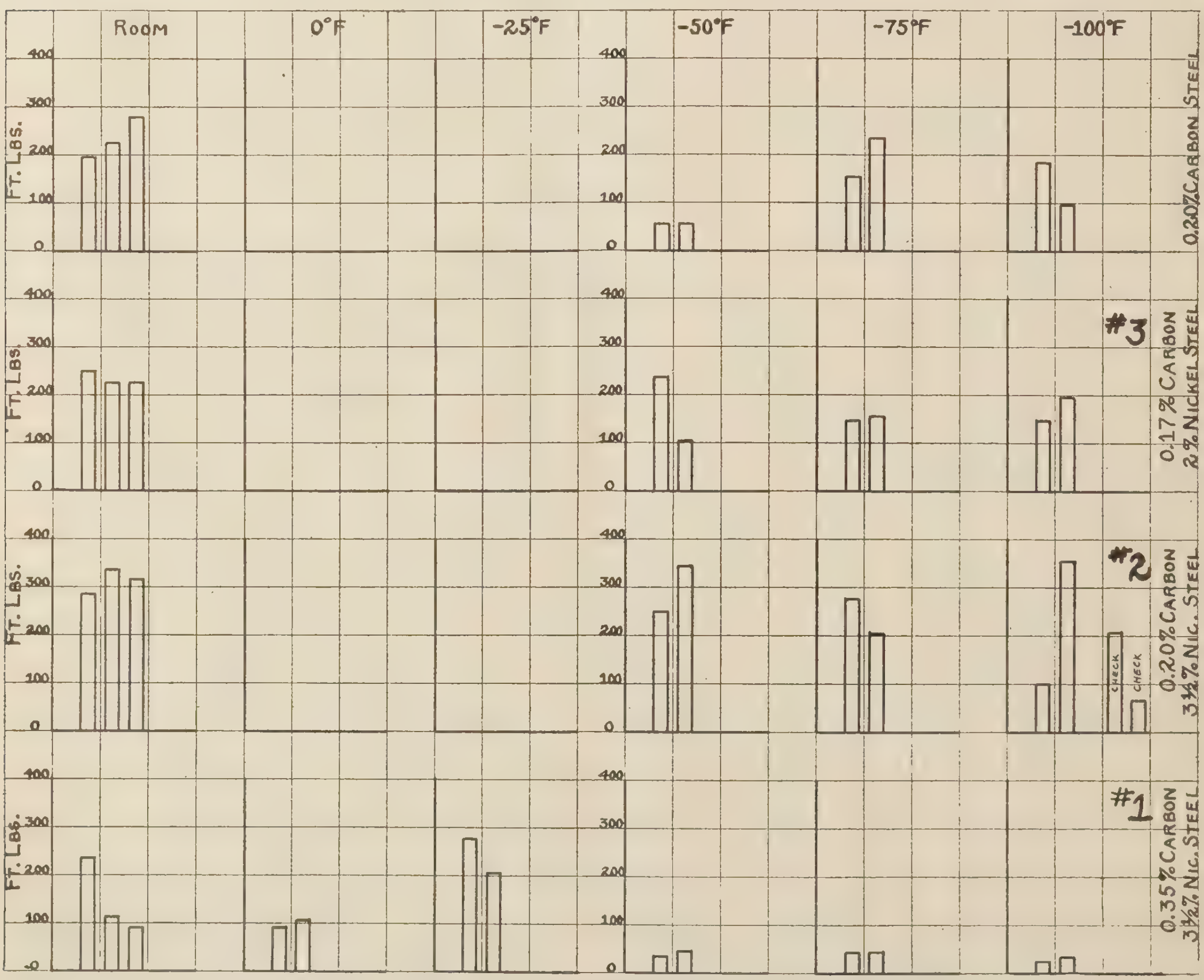


Fig. 15 Results of Low-Temperature Tensile Impact Tests on Butt Welds on $^1/_2$-In. Plain-Carbon and Structural Nickel-Steel Plate Stress Relieved at 600 C

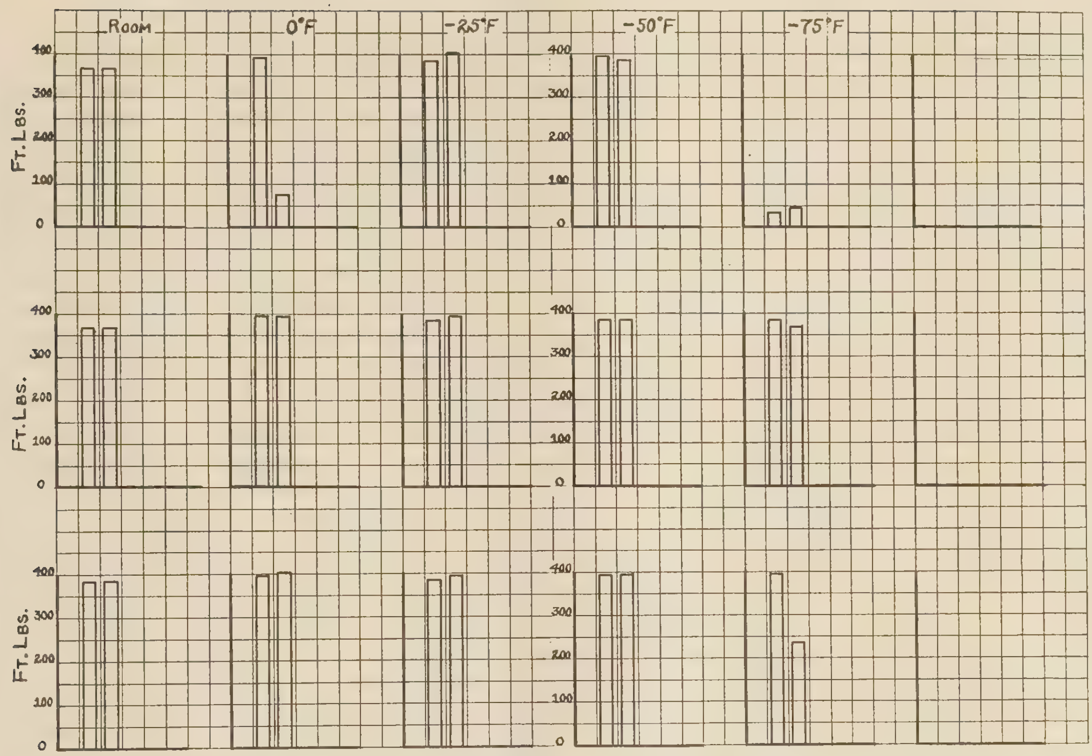


Fig. 16 Results of Low-Temperature Tensile Impact Tests of Unwelded 1/2-In. Structural Alloy-Steel Plate Stress Relieved at 600 C

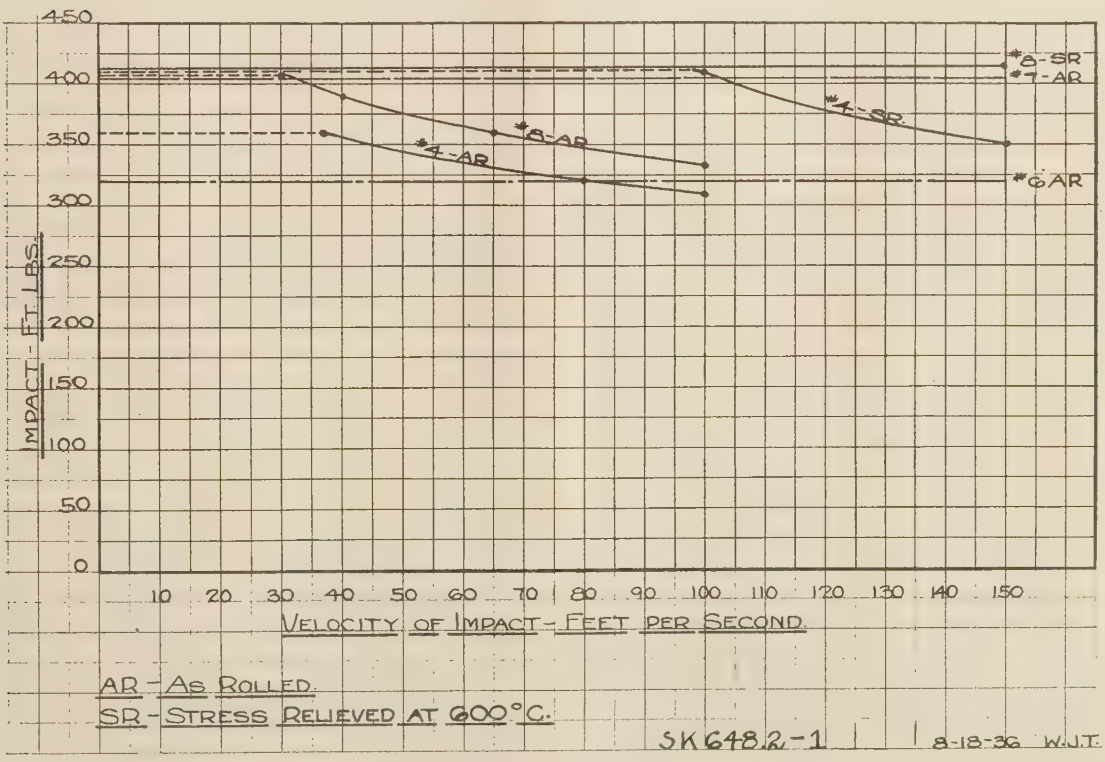


Fig. 17 Results of High-Velocity Tensile Impact Tests of Unwelded 1/2-In. Structural Alloy-Steel Plate
(Numbers on curves refer to steels listed in Table 1.)

TABLE TENTATIVE ORDER OF ACCEPTABILITY OF STEELS FOR WELDING PURPOSES

Strength of plate	Plate number[a] As rolled	Plate number[a] Stress relieved
Tensile strength, lb per sq in.:		
95,000 and over	11 4 10 1	1 . .. ..
85,000 to 95,000	10 13 2 9 5 7	11 10 4 13 . .
75,000 to 85,000	5 12 7 3 8	12 2 7 9 ..
60,000 to 75,000	6 8	3 8 5 6
Proportional (elastic) limit, lb per sq in.:		
55,000 and over	4 1	4 2 1 7
45,000 to 55,000	7 2 6 1 9 ..	7 5 3 13 11 8
40,000 to 45,000	13 8 3 9 5	6 8 .. .. ..
35,000 to 40,000	3 5 12 11 10	12 9 10 .. ..

[a] Numbers refer to Tables 1 and 2.

be understood, however, that these data cover only the phases of weldability discussed here and, that other factors not studied here may change the desirability of any given composition of steel for a given welded structure.

The specified minimum tensile-strength and yield-point requirements for steel plate used in welded ordnance structures are 85,000 and 50,000 lb per sq in., respectively. These limits allow steels Nos. 11, 1, 13, 10, and 4 to be considered for tensile strength in the stress-relieved condition, while steels Nos. 4, 1 2, and 7 can be considered for elastic limit in the stress-relieved condition. Steels Nos. 5–3–13 are on the line at an elastic limit of 50,000 lb per sq in., but this is too close to the minimum to be considered.

This leaves steel No. 1, as second choice in both cases because of the high hardness in the heat-affected zone. Steel No. 11 is best for tensile strength and steel No. 4 is best for elastic strength and both are of considerably lower carbon content than steel No. 1, which makes for lower hardness in the heat-affected zone, or better weldability, although steel No. 11 falls below the limit on elastic strength. Steel No. 2 (0.22 per cent carbon and 3.5 per cent nickel) is very acceptable on elastic strength but falls down below the limit of 85,000 lb per sq in. tensile strength, and its weldability (heat effect) is on a par with steel No. 4, which with steel No. 1 is included in both groups.

It is hoped that eventually a similar study may be made of weld properties on these steels when the welding tests have been completed. Under those conditions comparisons may be somewhat complicated by differences in electrode materials.

The tensile properties of steels as normally determined present a relationship between certain characteristics of the materials which may change considerably under other types of tests, such as low-temperature impact and high-velocity impact.

In Fig. 14 is shown the shape of test specimen used at the Watertown arsenal for low-temperature impact tests of butt welds. The specimen shown is used for $^1/_2$-in. thick plate and has a straight test section 1 in. long with $^1/_8$-in. radius at each end, or a total length between shoulders of 1.25 in. The cross section is $^1/_4$ in. wide by the plate thickness, and the total length of the specimen is approximately 3 in. Tests of the unwelded plate are made also with this specimen.

Low-temperature impact data on butt welds of $^1/_2$-in. structural carbon-steel and structural nickel-steel plate are shown in Fig. 15. The No. 2, 0.20 C-3.5 Ni steel holds up best at low temperature, and the effect of the carbon on the subzero impact strength is clearly indicated by comparing the results for steels Nos. 1 and 2. The structural carbon-steel plate shows a decided drop at —50 F.

Low-temperature impact data on plates Nos. 8, 9, and 10 listed in Table 1 are shown in Fig. 16. The test specimen used was that shown in Fig. 14. Steel 9, containing high manganese with medium carbon, appears to hold up better at low temperature under impact loads than either No. 8 or No. 10.

In Fig. 17 are shown data on four structural alloy steels tested in the variable-speed impact testing machine at the Watertown arsenal laboratory. The data shown indicate that there is no drop in the impact strength of the copper-nickel steel in the as-rolled condition up to 150 fps velocity of impact. The high-phosphorus steel shows no drop up to this velocity in the stress-relieved condition but as-rolled this steel does show a drop at a velocity of about 30 fps. These data indicate the value of stress relieving.

Steel No. 4, also included in Fig. 17, shows not only an increase of critical velocity on stress relieving but also an increase of impact strength of approximately 15 per cent.

At the time this paper was prepared these tests were not completed but it is hoped that eventually the data on low-temperature impact and high-velocity impact will become available not only on the plate materials themselves, but also on butt welds of these materials made with various electrodes. With data of this kind available, it is quite probable that the comparison of weldability may be somewhat different from that derived by the tensile test alone.

It is of interest to consider, for a moment, the possibilities offered for comparing structural steels by the tests mentioned in this discussion. The tensile test gives values which the designer can use for design of structures for static loads. The tensile impact test[3] either at room or low temperatures does not give values which the designer can apply directly to his design, but the results of such tests are a valuable guide to the selection of material for the design where dynamic loading is to be encountered.

In conclusion, the author wishes to acknowledge the invaluable assistance and many helpful suggestions of Colonel G. F. Jenks, Commanding Officer, Watertown arsenal, and Chairman, Subcommittee on Industrial Research, Engineering Foundation. The help given by the International Nickel Company, and the Climax Molybdenum Company, and the privilege of using data obtained under their sponsorship is appreciated. The samples of plate material furnished by the Carnegie Steel Company, the Republic Steel Corporation, and the Youngstown Sheet and Tube Company have greatly facilitated this study of the welding quality of low-alloy steels.

[3] "The Relation Between the Tension Static and Dynamic Tests," by H. C. Mann, Proceedings A.S.T.M., vol. 35, part 2, 1935, pp. 323–335.

Rolled Steel in Machine Construction

By H. G. Marsh,[1] Pittsburgh, Pa.

The author discusses the extent to which welding is used in the design and construction of machine tools, and advances reasons why its application is not more universal.

Last year 80,000,000 lb of electric-welding rods were used. On the basis of 25 to 40 lb of welding rod per ton of steel, it indicates that between 2,000,000 and 3,000,000 tons of steel were welded. Automobiles, buses, street cars, high-speed trains, and airplanes all depend to some extent on the satisfactory performance of welds. Welding has become so common that the public accepts it without comment, and in many cases it is actually preferred, especially where general appearance is an important factor.

The magazine *Machine Design* in the last year described 133 newly designed machines, 19 of which were made of welded rolled steel, 45 had been partially converted from castings to welded steel, and 69 were still made of castings. The machinery and equipment described were of all classes.

The *American Machinist* for the same period described 263 machine tools, of which 10 were made entirely of welded rolled steel; 14 were partially converted, and 239 were made entirely of castings.

In the field of power shovels and road-building equipment, it has been estimated that 50 to 60 per cent of the weight of all the machinery built has been converted from castings to rolled steel. In other lines, such as presses, brakes, electric cranes, and heavy industrial equipment, the conversion has been from 15 to 50 per cent completed. The large electrical manufacturing companies have gone to the economical limit set by present welding practice.

Notwithstanding all this progress, the use of welding in general machine construction is still much restricted, as compared to its ultimate possibilities. There is much to be done before the benefit of this type of construction will be realized to the proper extent. Those familiar with this development and its possibilities estimate that the conversion from castings to welded rolled steel for *machinery in general* is only about 15 per cent complete. The reason for the apparent backwardness of welded steel in this field is not because steel is an unsuitable material for machine construction. On the contrary, it has all the qualities required for the purpose. It is low in cost, reliable, and available in a great variety of shapes and qualities. Nor is it a matter of cost, for in most cases where welded construction has been adopted the costs have been reduced. Even if such were not the case, the use of steel would not necessarily be prohibited, for if it does a better job and meets with the acceptance of the buyer, it can command a higher price. Steel has replaced wood, cast iron, malleable and steel castings in hundreds of cases at increased cost but its suitability for the purpose made its use advisable.

The hesitant progress of welded rolled steel in the machinery field is due to a readily discernible apathy toward welding on the part of many engineers. There are a large number who still regard welding as just a handy tool for the millwright in case something breaks. They hesitate to use welding in the construction of machinery they design, or to advocate its use in the machines they purchase, because they are not sufficiently informed of its possibilities and demonstrated performance. A little more familiarity with the process of welding would dispel much of this doubt.

The manufacturer of a certain type of machine, which is built entirely of welded steel, recently stated that they were very careful to eliminate all evidence of welding. No reference to the machine's being welded is made in any of their advertisements or other printed matter. This is done because of the sales resistance to welding they have experienced in marketing their product. The machine is not sold to the general public, or this procedure would be unnecessary. It is sold to engineers and production managers, who, above every one else, should know about welding and have faith in it when properly used.

Another manufacturer recently went to considerable expense to duplicate exactly in welded steel a standard machine of cast construction so that a service test could be secured. All evidence of welding was carefully concealed so there would be no objection on the part of any one in the buyers' operating organization because of its being welded.

It was very noticeable at a recent machine show that manufacturers who used welded construction took pains to conceal it just as much as possible. This was not solely for the sake of appearance, although it is admitted that a smooth well-rounded fillet has a very definite appearance value. A gentleman who was much interested in the subject at the time called attention to the fact that regardless of all the publicity on welded construction, he saw none of it at the show. Fortunately, it was possible to give him the names of about twenty well-known machine-tool manufacturers who were exhibiting welded structures there, but unless one was looking for this feature particularly, it was rather difficult to discover many examples of it.

The steel industry's interest in the market now being expanded by welding is not an entirely selfish one. Steel is a progressive industry and while naturally interested in extending the use of its product, it is also interested in developing its equipment along the best economical and engineering lines. It realizes the advantages of this construction and uses it wherever applicable. It is favored and many times specified in the equipment bought. With this twofold interest in the development of welded construction, the steel industry is watching its progress very carefully.

Sales resistance to welded construction exists among many engineers because they have not employed it to a sufficient extent to become acquainted with its various methods of application and the results obtainable thereby. They do not use it in their designs because they believe that only a welding technician can design welded structures. This is an erroneous point of view, especially with respect to the type of structures under discussion, i.e., machinery.

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[1] In charge of sales service and training, Carnegie-Illinois Steel Corporation, Pittsburgh, Pa. Mr. Marsh was graduated from the mechanical-engineering course, University of Pittsburgh, in 1906. Shortly afterward he entered the employ of the Carnegie Steel Company in the operating department, serving successively as sales engineer and assistant manager of the speciality sales division before assuming his present duties. He has been interested for a number of years in welding from the promotional point of view.

Contributed for presentation at the Welding-Practice Symposium sponsored jointly by The American Society of Mechanical Engineers and The American Welding Society, to be held at Cleveland, Ohio, October 22 and 23, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

There is nothing mysterious about a weld. It is simply a joint by which two pieces of metal are held rigidly together. Rivets and bolts perform the same function. If the physical qualities of the weld, such as tensile strength, ductility, and resistance to fatigue and impact are known and these fulfill the requirements, the joined pieces may be considered as one piece, and the fact that they are welded may be entirely forgotten.

The question is immediately asked: "What if the weld is not perfect?" It does not have to be perfect. Welds generally consist of a number of layers of weld metal superimposed on each other and fused together, and the possibility of slight imperfections in several layers occurring at the same point is very remote. It is no more necessary to examine a weld for blow holes, segregation, and cracks than it is a casting. Nor is it more necessary for a welded joint to be perfect than a riveted one, because conventional factors of safety take care of discrepancies or variations in workmanship and material.

The question as to whether there is a standard of workmanship among welding operators, which may be depended upon, may be answered in the affirmative. This is especially true in the field of machine construction. It is assumed that the designer of such machinery will depend upon a competent welding unit in his own organization, or that the work will be sent to a reputable commercial weldery. In either case, the standard of the work done will be sufficiently high and uniform for his purpose. With modern apparatus and materials, welding of mild steel presents no difficulties, and resulting welds are quite satisfactory from every viewpoint.

It is realized that special conditions and materials modify welding procedure and introduce problems which require a specialized knowledge of welding. Some of these involve metallurgy and heat-treatment, and must necessarily be left to experts in this line. Expert advice is readily obtainable when such special problems are encountered.

Many detailers design riveted joints with little knowledge of riveting except that given in our standard engineering handbooks. They seem to have learned the difference between a rivet in shear and one in tension without any exhaustive study of the subject. Give them the same data on welded joints, and a general knowledge of the properties of weld metal to be added to their usual good mechanical sense, and they will design welded structures in a very acceptable manner. How disastrous it would be if drawings of riveted structures were sent to the shop marked "rivet here" and the mechanic was left to exercise his own judgment as to the size and spacing of the rivets. For a while this was and still is the accepted method of "detailing" welds in some shops.

It is an exploded theory that a welding technician is required for the design of welded machinery. With standardized data on welds comparable to those available on riveting, a designer of presses, who has spent years in that line will design a better welded press than the best welding technician in the world. In the Pittsburgh district there are several hundred men turning out creditable designs without a first-hand knowledge of the technical details of welding.

The way to break down the existing sales resistance to welded construction is to start the thousands of designers in this country using it. The way to start them using it is to give them simple and reliable data on welds. This is a challenge to all who are interested in the more widespread application of the method, and every proper means should be used to place the necessary information in the hands of those who can make effective use of it.

The designer is of utmost importance to the development of industry. He does not have time to become a welding technician, nor is it necessary. Supply him with reliable and workable data, and with his keen insight and ingenuity he will quickly grasp the principles of welding and lift welded machine design to new heights.

Welding Heavy Machinery

By C. A. Wills[1] and F. L. Lindemuth,[2] Youngstown, Ohio

This paper is a presentation of the problems met with in the designing, fabrication, and stress relieving of welded machine parts and other types of welded equipment. In discussing the solutions of these problems, the authors refer to the welding of fabricated machine parts and equipment as accomplished by the company with which they are associated.

In MANUFACTURING or fabricating any kind of equipment or part thereof, obviously the first thing that must be considered is the design. There will generally be some more or less fixed overall and detail dimensions and clearances which must be adhered to and certain requirements as to strength in the various parts of the structure. In a machine there will be a combination of such parts as wheels, shafts, bearings, links, cylinders, and rods supported in a frame which will often be of welded steel. Quite often the machine under consideration is similar to a machine which has been previously made, but the parts now to be made of welded steel had previously been made of some other material.

Various methods of construction have certain limitations, and the product made under such limitations is not ideal although it is the best that can be made with the construction method used. For instance, in riveted construction awkward expedients are used sometimes merely to provide sufficient clearance to insert and drive the rivets, excess material is used to provide connections of sufficient strength, and a large number of auxiliary members, such as splice plates and clip angles are added.

In making castings, the limitations imposed are those of the casting process, some of which are: (1) The casting should be fairly uniform in thickness throughout so that the metal will fill all parts of the mold and thus cool at a fairly uniform rate. (2) The shape is limited by what can be economically molded. (3) It may be necessary to make a large or intricate member in several pieces which are machined and keyed and bolted together. (4) A casting is necessarily one kind of material.

However, intricate and irregular shapes can be produced much more advantageously by casting than they can by any other method of construction, especially if more than one casting is to be made from an expensive pattern.

When a welded-steel structure is being designed, the designer must realize what can or cannot be done in welded steel in order not to handicap the design from the start with the limitations of some other construction process. About the only limits as to size and shape of a welded-steel structure are those imposed by machining and shipping facilities: thick and thin members can be welded together; strong, rigid sections, such as box-shaped sections, difficult even to approximate in castings, are easy to make; and various materials can be combined in one piece. By considering the welded job in the light of the possibilities and limitations of the welding process, and not as a means of producing an imitation of some other kind of construction, better results are obtained from the point of view of strength, rigidity, cost, and appearance. For example, a casting may be an I-section with stiffening ribs, which is only a fair design for strength and rigidity, and may have an outside surface with a series of unsightly pockets which are hard to keep clean. With welded steel it may be possible to make this same surface a box section with straight and smooth surfaces which can be kept clean with a minimum of care and is pleasing in appearance. A comparison of welded construction and riveted construction of a ladle is shown in Fig. 1 while a comparison of a welded and a cast machine tool is shown in Fig. 2. There may be certain parts of the construction which become too complicated to work out economically in welded steel, but which are easily made in a casting. For these parts a small steel casting can be made and welded into the large welded-steel member. This type of construction is illustrated in Figs. 3 and 4.

The thickness of the material to be used throughout the structure and the amount of welding necessary can be determined from an analysis similar to that made by a structural designer in determining the sizes of various members and the joints. The authors have no fixed rule that certain thicknesses of plate require welds of a certain size or shape. They sometimes find that a certain thickness of plate may be sufficient to carry the loads but not sufficient to enable the welder to build up the required shape and size of weld. In a case like this the thickness of the plate may be increased or the design may be changed to meet the welding requirements.

In designing for rigidity rather than strength, it should be kept in mind that the modulus of elasticity of steel is twice that of cast iron, that is, it is twice as stiff.

Too much welding can have disadvantages. The deposited weld metal is a very expensive material, costing from $0.50 to $1 or more a pound, based on the rod used to make the weld. Thus, excessive welding runs up the cost with no compensating advantage. As the weld metal cools it tends to contract, so that the whole structure assumes a distorted shape, and the more weld metal there is the greater this distortion will be. The stresses in these welds tend to relieve themselves with age and the whole structure goes back to its original undistorted shape. In a structure, in which for some reason the welding stresses are not relieved and in which accurate permanent shape is necessary, an excessive amount of welding can be of considerable disadvantage.

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[2] Chief Engineer, The William B. Pollock Company. Mem. A.S.M.E. Mr. Lindemuth was graduated from the Pennsylvania State College in 1907 with the degree of B.S. and received his M.E. degree from the same college in 1913. In 1919 he was associated with Perin and Marshall, consulting engineers, New York, N. Y., and while in their employ acted successively as assistant general manager, general superintendent of construction, and chief works engineer of the Tata Iron and Steel Company of India. Afterward he was chief engineer of the Colorado Fuel and Iron Company, Pueblo, Colorado, and engineer with the Mesta Machine Company, Pittsburgh, Pa. He has been employed in his present capacity since 1932.

Contributed for presentation at the Welding-Practice Symposium sponsored jointly by The American Society of Mechanical Engineers and The American Welding Society, and held at Cleveland, Ohio, October 22 and 23, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

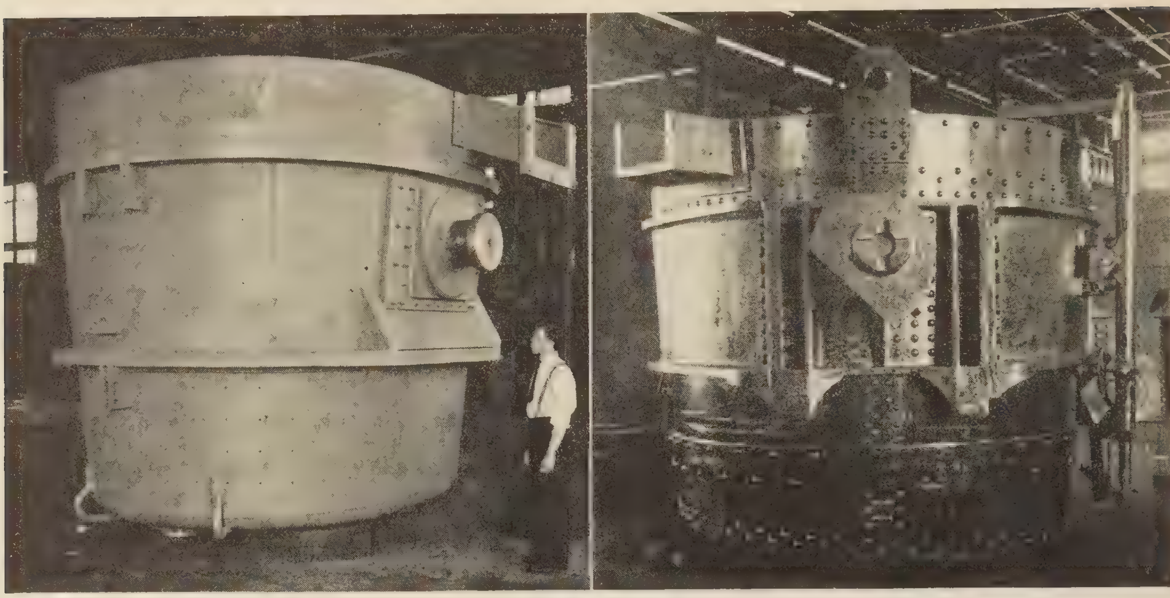

FIG. 1 WELDED AND RIVETED STEEL-MILL LADLES

(The two ladles are approximately the same size but the welded ladle is 33 per cent lighter than the riveted construction. The pads of the welded ladle into which the trunnions are pressed are highly stressed and there was not room to develop sufficient welding. The necessary strength was obtained by a combination of welding and riveting. The welding stresses were relieved at approximately 1200 F to obtain maximum strength.)

Occasionally in highly stressed structures there may be difficulty in getting the necessary amount of welding in the space available. There are various ways of circumventing this difficulty. The amount of welding along the edge of the plate can be increased by serrating the edge. Another scheme is to plug weld, which is punching or drilling holes through one plate of an overlapping pair and filling the holes with weld metal. These welds are rather uncertain and are not to be recommended because it is difficult to obtain adequate penetration at the bottom and between successive layers of weld, and to work out the slag as the welding proceeds when holes are small and metal is thick. A combination of welding and rivets and fitted pins or bolts can be used. Strenuous objections have been made to the use of this type of joint on the basis that it is not theoretically correct, which is true. However, a riveted joint with three or more rivets in line is just as incorrect, and yet we do know that it is satisfactory in spite of theory. Very few joints other than a welded butt joint will stand a rigid theoretical analysis.

Sometimes it is desirable to use two or more kinds of material in one structure, as, for example, in the case of the gear blank shown in Fig. 5 where high-carbon or alloy steel is used in combination with low-carbon steel. Such combinations are impossible in a casting. However, the structure can be made either of a compromise material or separate pieces of the different materials fastened together in some more or less expensive manner.

But with welding, various materials can be combined into one piece, the only limitations being to find a welding rod which can be used satisfactorily to bond the various materials and to use materials which have sufficiently similar properties, such as modulus of elasticity or coefficient of expansion, for the service for which they are intended.

When the authors have the alternative of using either light sections reinforced with ribs, gussets and knee braces, or heavier sections which do not require these small braces, the heavier sections are generally preferred. Such little braces are expensive, it is hard to make the welds around them tidy, and they spoil the appearance of what would otherwise be large clean surfaces.

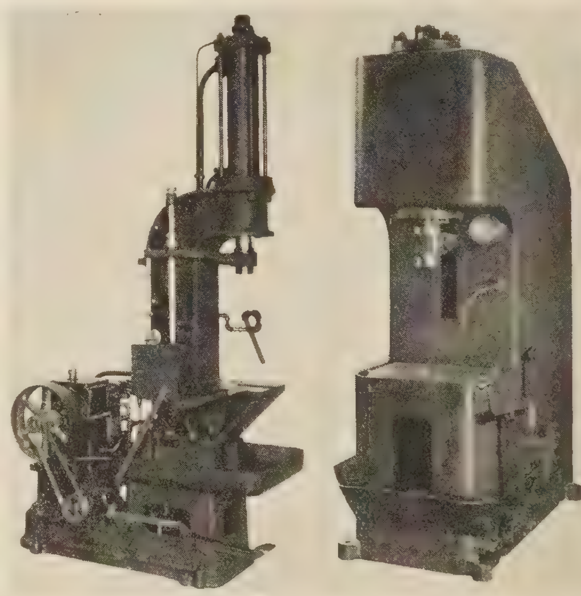

FIG. 2 WELDED AND CAST MACHINE TOOLS OF THE SAME TYPE
(Courtesy of The Oilgear Company Milwaukee.)

The authors do not advise the building up, peening, chipping or grinding of welds to make them look like, say, large fillets on a casting. This is a matter of individual taste, but the authors believe that a well-made weld has sufficient beauty of its own and that it is not necessary to try to make it look like something else. If the weld is not chipped or peened the inspector has a better opportunity for a surface examination, which is often the only inspection possible to determine the workmanship of a finished weld. Peening to smooth the weld may temporarily cover up certain defects such as porosity and unevenness. When not skillfully done it may mutilate the weld to a point where its

strength is seriously impaired. On the other hand, there are certain kinds of work where the natural roughness in the surface of the weld is not desirable, in which case the proper thing to do is to grind, chip, or peen the welds to get the appearance desired, or the surfaces can be smoothed with filler before the final painting.

In designing and fabricating welded work the authors consider only the possibilities and limitations of the welding process and do

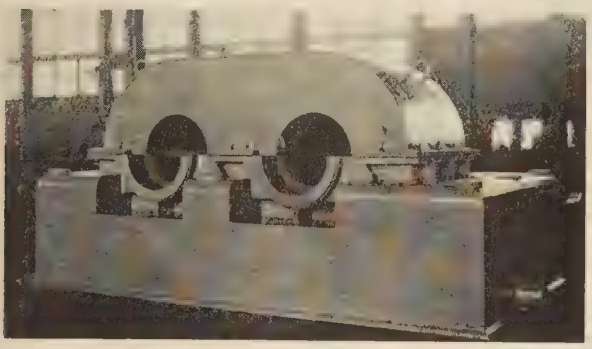

Fig. 3 Welded Gear Case the Intricately Shaped Bearings of Which Are Steel Castings Welded Into the Rolled-Steel Frame

(The main frame of the gear case was annealed at 1200 F to relieve stresses and to prevent distortion during and after machining. The cover is of thin plates and was not annealed, but the design was worked out so that slight distortion of the cover would not affect the alignment of the principal members.)

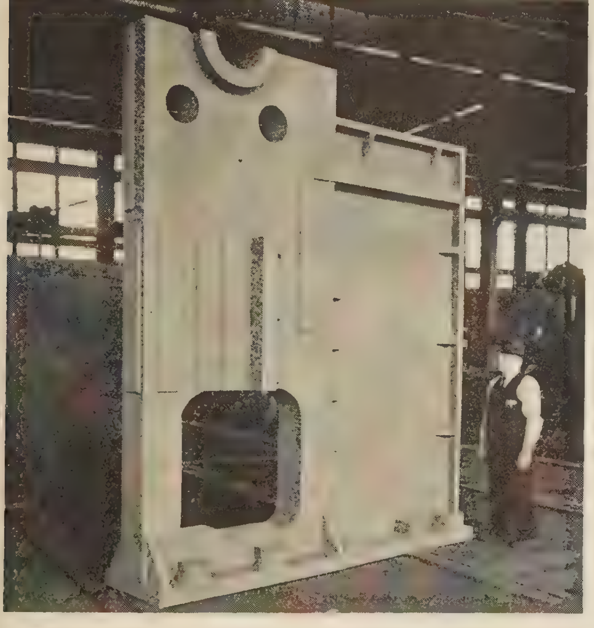

Fig. 4 Frame for Shear Made of Heavy Rolled Plates With the Top-Bearing Steel Casting Welded Into Place

not limit the size, shape, and appearance of a structure to what can be made by another process. In this way it is possible not only to get the most economical results but often to get better strength and appearance than can be obtained in any other way. This is illustrated in Figs. 6 and 7.

Stress Relieving Welded Structures

The question of relieving the locked-up stresses in the welds is one in which there are great differences of opinion as to necessity, desirability, and methods for accomplishing it.

As stated before, weld metal tends to decrease in volume on cooling and is restrained from doing so by the parent metal. This sets up stresses both in the weld and the parent metal and pulls the entire structure out of shape until an equilibrium of stresses is established throughout the structure. These stresses decrease and disappear in time in a manner similar to the way the stresses in castings are relieved by aging. As the stresses decrease the shape of the structure changes until the strains finally disappear.

When a welded structure in a state of strain is machined and some of the metal is taken away, the previous equilibrium is destroyed and a new one set up, accompanied by a change in shape of the entire structure, so that it may be impossible to machine such a piece satisfactorily.

When weld metal is in a state of strain its efficiency as a joining material is impaired.

Fig. 5 Welded Gear Blank With Hubs of Forged Steel, Web Members of 0.20 Carbon-Steel Plate, and Rim of 0.45 Carbon-Steel Plate

When accurate dimensions or maximum strength is necessary, the internal stresses in the weld metal should be removed in some manner before any machine work is done. These stresses can be removed in two ways: (1) By cold working or peening the weld metal to decrease its volume to what it would be if there were no strains in it, and (2) by heating it to a sufficiently high temperature for a sufficient length of time, which is simply an accelerated aging process.

To relieve the stresses by heating, the terms "annealing" and "normalizing" are used occasionally without further explanation. "Annealing" is a general term for a number of heating and cooling operations through a wide range of temperatures to accomplish a variety of purposes, one of which does happen to be the relieving of stresses. "Normalizing" is a more definite term which has little, if anything, to do with relieving the stresses. In time we will probably arrive at some better agreement as to how stresses should be relieved and coin some terms to describe the processes. Most often when the term "annealing" is used it is intended that the stresses in the welding should be relieved by heating. Oc-

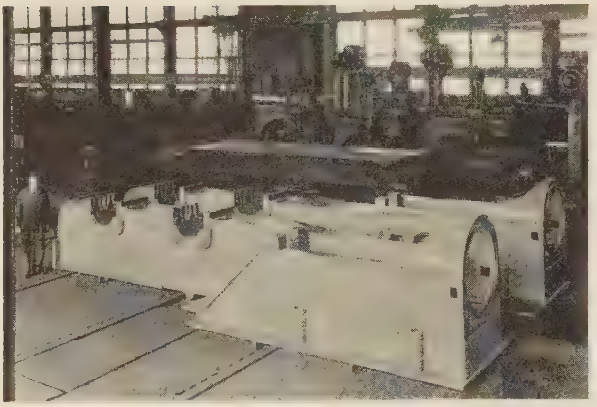

Fig. 6 Blooming-Mill Engine Frame

(The welded frame was fabricated to replace castings which had broken and for which expensive patterns were no longer in existence. The principal members of the frame are box sections designed to give maximum strength. The bearings are castings welded into the rolled-steel plate. The frame was annealed at 1200 F.)

casionally, annealing is required for other reasons, such as to produce a definite microstructure in the steel, in which case specific directions for annealing should be given because each reason for annealing requires a treatment different from that used to relieve the stresses.

The peening of welds is an uncertain method of relieving stresses because it is difficult to tell just when the right amount of peening has been done. If the weld is not peened sufficiently the effect that is desired is not obtained; if it is overdone, tension stresses in the weld metal are simply changed to compression stresses, and in addition the weld metal may have been damaged. The process is useful, however, because there are often times where it is impractical or impossible to relieve the stresses in any other manner, but when it is used its limitations and uncertainties should be kept in mind.

The safest procedure is to heat the welded object in a furnace to a sufficiently high temperature for a sufficient length of time. For machinery or similar parts the details of the heating process can hardly be expressed by a simple rigid formula. In the A.S.M.E. and A.P.I. codes for pressure vessels the rule given is to heat the object to about 1200 F and hold it at this temperature for one hour for each inch of thickness. The heat is to be raised at the rate of not over 400 F per hr divided by the thickness of the metal in inches, in no case faster than 400 F per hr; and the cooling is to be done no faster than 500 F per hr divided by the metal thickness in inches, and in no case more than 500 F per hr. The vessels for which these very definite rules have been formulated are of the same kind of material, low-carbon steel. The welds are solid continuous welds through the metal, and the metal throughout the entire structure has little variation in thickness and, therefore, requires practically uniform treatment.

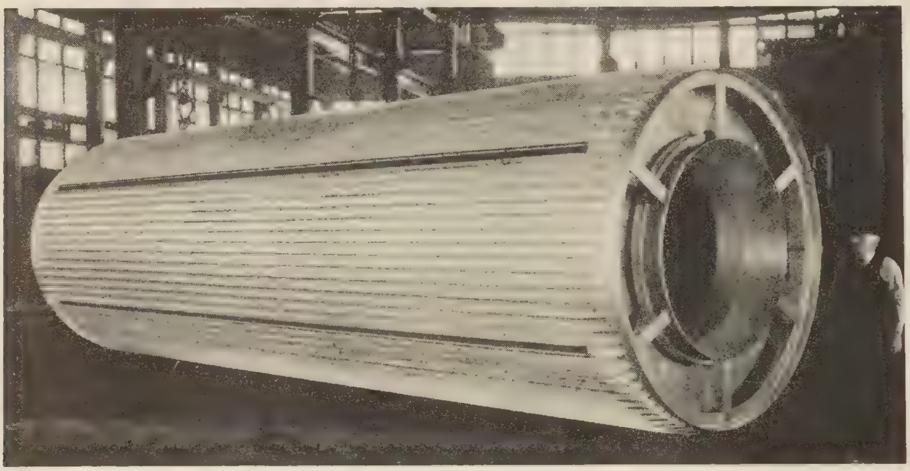

Fig. 7 Part of an Electric Precipitator Made From Thin Smooth Sections to Accurate Dimensions

A welded machine part may be of almost any size or shape, several materials of a variety of metallurgical natures may be welded together in one structure, and there may be a wide ratio of thickness between the thinnest and thickest sections. It can, therefore, be seen that the stress-relieving treatment for any individual piece, especially of the type shown in Fig. 8, must be a matter of experience and good judgment. For example, if a casting 6 in. thick is welded to plates 0.75 in. thick with welds 0.75 in. deep, it should not be necessary to heat the object at a rate of less than 100 F per hr and hold it in the furnace at 1200 F for 6 hr in accordance with the pressure-vessel formula. It should be brought up to heat more slowly than would be permissible if all the sections were 0.75 in. thick, to allow the heat to penetrate the thick casting without overheating the thin plate, and held at high temperature for a longer time than would ordinarily be required for welds 0.75 in. deep.

Trouble is often encountered when relieving the stresses. Large thin plates without stiffeners along the edges attached to heavier members bend during welding, and then during heat-treatment they buckle before the heavier members are hot due to the heat and locked-up stresses. Other large pieces may be of such size and shape that no matter how they are supported in the furnace they may get out of shape during the heating and cooling

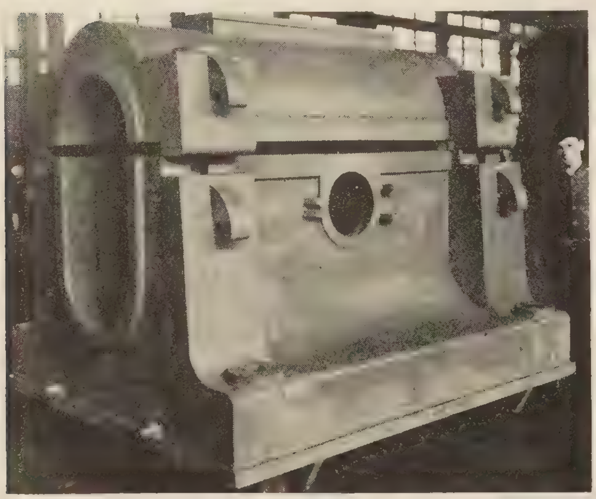

Fig. 8 Rolled-Steel Pinion Housing Annealed for Strength to Prevent Distortion During Machining

process and come out in worse condition than when they go in. Straightening is difficult and it is often advisable to permit the structure to remain as it comes from the furnace. Sometimes the structure assumes such a distorted shape even with the most careful treatment that it must be straightened. Experience has shown that such problems can be overcome only by properly designing the structure in the first place with permanent or temporary stiffeners, or by changing the shape so that the effect of the welding stresses will be a minimum, or by changing the amount of welding. In changing the amount of welding, the minimum permissible amount is deposited at one point while more than is actually required for strength is deposited at another point in order to balance the stresses throughout the structure so that its shape will not change appreciably during fabrication and heat-treating.

Sometimes the metallurgical structure of some of the materials used in a welded fabricated structure becomes damaged in forming or welding it. When this object is placed in the furnace, the problem is not only that of relieving the welding stresses, but also of furnishing additional heat-treatment to obtain the correct metallurgical structure of the material.

For example, a piece of 0.20 carbon steel is harmed little, if any, by hot or cold forming or by the welding process, but a piece of 0.50 carbon steel may be seriously damaged metallurgically in both the forming and welding. Heating to 1200 F would relieve the welding stresses in either piece, but to correct metallurgical changes in the 0.50 carbon piece it may be necessary to normalize it at 1500 F instead of annealing at 1200 F. For stress relieving only it is best to keep the temperature around 1200 F. Steel under about 0.30 carbon heated to 1200 F will machine satisfactorily and there will be no scale on the surface, but if heated to much over this it will not machine so smoothly and the surface will be covered with a heavy scale which is expensive to clean off and spoils its appearance.

Except for the simplest welding jobs no general formula for relieving the welding stresses is possible. The welding stresses will relieve themselves in time at ordinary temperatures. Heating will accelerate this process. At 1200 F, the time required to relieve these stresses is measured in hours. Heat-treatment in excess of 1200 F, if it is not necessary for some other reason than simply for relieving the stresses, may do more harm than good.

Applications of Copper-Alloy Welding

By I. T. HOOK,[1] WATERBURY, CONN.

The author presents a brief survey of the uses of copper alloys as welding materials in industrial applications. In general, the survey includes three distinct fields: (*a*) Joining of copper alloys by various brazing and welding methods; (*b*) joining of ferrous and other metals by copper-alloy weld metal; and (*c*) building up wear- and corrosion-resisting surfaces of copper alloys. The discussion is confined principally to the use of copper-alloy welding. Several of the applications cited are taken from published reports of different investigators, and references are given where details involved in copper-alloy welding may be found.

1 COPPER-ALLOY PIPE WELDING

One of the most familiar applications of copper alloys is the joining of copper and brass pipe. Much of this type of pipe in the thinner-walled sizes is connected by mechanical compression fittings, soft soldering, or silver brazing. In this discussion, however, the author will confine himself to pipe

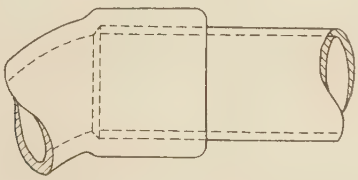

Fig. 1 Cast-Brass or Copper Recessed Fitting Sweat-Soldered or Brazed to Copper-Alloy Pipe

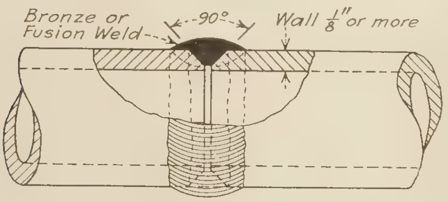

Fig. 2 Butt Weld in Copper or Brass Pipe

which is of sufficient thickness, usually 0.045 in. or thicker, to permit its being brazed or welded. The threaded fitting, still a popular method of joining brass pipe in the standard pipe sizes, is gradually giving way to one form or another of brazed or fusion-welded connection (1, 2).[2]

Figs. 1 to 4, inclusive, show, in longitudinal section, various forms of brazed or welded connections. The recessed-fitting connection shown in Fig. 1 is more often soft-soldered or silver-brazed but it may be Tobin bronze-brazed as well.

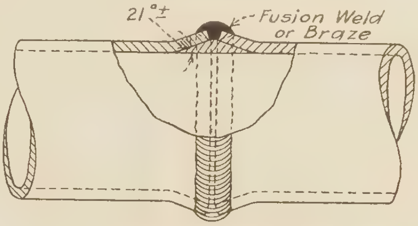

Fig. 3 Taper-Weld Line Connection

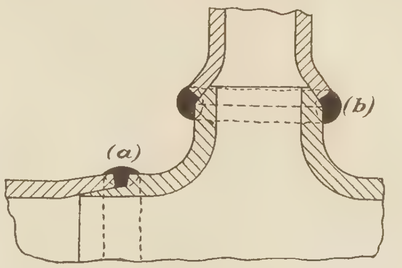

Fig. 4(*a*) Taper-Weld Connection to Fitting. (*b*) Short Bell-and-Spigot Line or Fitting Connection

Simple butt welds, illustrated in Fig. 2, in brass or copper pipe of $^1/_8$ in. or more wall thickness are commonly made with a yellow-bronze welding rod. In the chemical industry, welds in copper pipe are, for purposes of uniform corrosion resistance, usually fusion welds with a deoxidized-copper welding rod.

Where the requisite skill in the welding operator is available, the installation may be made with connections as shown in Fig. 2. However, owing to the fluidity of the molten copper alloys and the high heat intensity (for copper pipe) required, the author prefers the use of the taper-weld connection (3, 4) shown in Fig. 3 and Fig. 4*a*, or the short bell and spigot shown in Fig. 4*b* (1, 2, 5, 7). The taper-weld connection is designed for any size of pipe for any wall from the thinnest to the thickest. It may be used with the silver brazing alloys as a sweated connection, with Tobin bronze, or as a fusion weld. So used, it acts as a back-up, forestalling the dropping of weld metal inside the tube and the melting of holes in the tube wall.

Since nearly all of the commercial copper pipe and tubing is made of phosphorous deoxidized copper, there is no content of cuprous oxide to give trouble in the welding operations. Whereever it is possible to use diverse metals, it will be found that soft solder, silver solder, or a yellow bronze such as Tobin will be easy to apply. But where the weld metal must exactly match the base metal in its chemical properties, a deoxidized-copper welding rod can be used.

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Contributed for presentation at the Welding Practice Symposium sponsored jointly by The American Society of Mechanical Engineers and The American Welding Society, held at Cleveland Ohio, October 22 and 23, 1936.

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[2] Numbers in parentheses refer to Bibliography at the end of the paper.

Brazed or Welded Copper Vessels

One application of welded copper of particular interest to mechanical engineers is its use in locomotive fireboxes. We see little of such usage in this country but in Europe, the British Isles, Asia, Africa, and Australia, the copper firebox is the rule rather than the exception. The application is briefly described here as many of the lessons concerning the use of welded copper learned in this severe service are useful to copper fabricators in this country. These large sheets of arsenical, tough-pitch copper have generally been of riveted construction though repairs on them have been made by fusion welding since 1925 or earlier. One objection to riveted copper pressure vessels lies in the difficulty in keeping the joint tight. Copper cannot be caulked as effectively as can steel and a leak does not corrode shut as usually happens with steel.

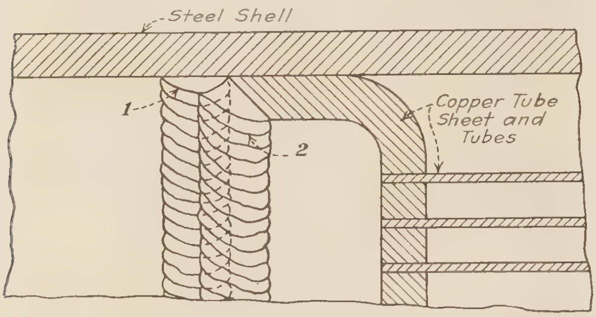

Fig. 5 Connection Between Deoxidized-Copper Tube Sheet and Steel Shell

A survey of the European situation with respect to copper fireboxes in 1931 (8) discloses the fact that nearly all locomotives in Italy had repairs on their copper fireboxes made by welding. Two locomotives with all-welded copper fireboxes were put into service on the Italian State Railways in 1927 and they were giving satisfactory service in November, 1931, after four years of service.

Several questions will arise as to the use and the welding of copper fireboxes. Just why steel is used almost universally in this country for locomotive fireboxes and copper almost as universally in Europe and Australia, the writer is not prepared to say. However, copper has about seven times the heat conductivity of firebox steel and keeps cleaner of corrosion scale than the latter. Again, oxygen as it is released from boiling water is much more active than is atmospheric oxygen or the same amount of oxygen in cold water. Copper resists the attack of this active oxygen much better than does steel. These advantages appear to outweigh the advantage of steel in point of strength and initial cheapness in the minds of the European builders.

Again, the question will undoubtedly be raised as to why metallurgists and welders generally found copper so difficult to weld, and why welding procedures for it were not established earlier. Without going into detail, we may say that the high heat conductivity of the copper and the possible weakness arising from the cuprous oxide of tough-pitch copper were the two barriers to successful welding. These sources of difficulty occur in thin as well as thick copper, in copper vessels being fabricated for the chemical industry as well as for locomotive fireboxes.

In modern shops, welding equipment of ample heat capacity is now obtainable to overcome the high heat conductivity of copper sheet and to weld it using either the oxyacetylene torch or the carbon arc. The weakness arising from the cuprous oxide in the base metal can be avoided in three ways: (*a*) By using a sufficiently high welding speed (9) to avoid any change in the cuprous oxide, (*b*) by hot forging the weld or cold hammering and annealing it, or (*c*) by the use of deoxidized copper for the base metal.

Method (*a*) implies the use of the carbon arc. By no other means can a heat of sufficient intensity be obtained. A phosphor bronze fairly high in tin and phosphorous is the welding rod most frequently used in this process although some copper fabricators prefer Everdur for this purpose.

Method (*b*) is used to cause a redistribution of cuprous oxide after the weld has been made in tough-pitch copper by the oxyacetylene torch. The use of deoxidized copper (10) is the more logical method of overcoming any weakness arising from cuprous oxide no matter what method of welding or brazing is employed. In general, the corrosion resistance of deoxidized copper is identical with that of tough-pitch copper and its thermal conductivity slightly less.

Fig. 5 illustrates an interesting application of welding in the manufacture of a very efficient heat exchanger. A large bundle of copper tubes was silver brazed into a deoxidized-copper header. The header, instead of being clamped and gasketed between steel flanges as is customary in steam condensers, was welded to the steel shell in order to make a permanent seal.

An oxyacetylene weld using Tobin-bronze welding rod was one possibility. A gas-tight weld with a strength of approximately 45,000 lb per sq in. between bronze and steel was possible, thus developing the full strength of the soft copper. However, the gas weld was objected to because it would strain and loosen the silver-brazed connections of the copper tubes. Accordingly, it was decided to use the carbon arc and Everdur weld metal which would develop with both steel and copper a high-strength gas-tight bond.

One of the things which contributed to the success of this connection was the arrangement of the welding beads. In laying down bead 1, shown in Fig. 5, no attempt was made to catch the copper header, but a relatively low heat was used which would give the strongest weld metal and the best seal of the Everdur to the steel. Then bead 2 was laid down using a much higher heat which was necessary to overcome the high heat conductivity of the copper. This gave a strong fusion weld and gas-tight seal between the weld metal previously laid down and the deoxidized-copper header.

Welded Copper Alloys

Brass, the most common and, in many ways, the most useful of the copper alloys is employed in a great variety of ways with welded connections. Since the nickel silvers, the extruded brasses and architectural bronzes are also alloys of copper and zinc, they may be grouped with the brasses. Frequently, a welded connection is made for mechanical strength and gas or water tightness, and perhaps more frequently for an exact color match, the weld itself being ground flush with the base metal and polished. Architectural shapes, grills, ornamental facings, doors, window frames and sash, soda fountains and interior fixtures and trim, are often welded with the requirement that the weld be finished and inconspicuous. To do this, strips of the base metal may be used when a welding rod of the exact color match is not available.

Owing to its good temperature control and relatively wide-spreading heat, the oxyacetylene torch is almost always used. By suitable manipulation of the torch, the alloy can be welded without superheating the fused metal appreciably. Hence, the zinc volatilization can be kept to a minimum. Also the use of an oxidizing flame, which permits a film of oxide to form over the weld pool, helps in preventing the loss of zinc.

One of the welding rods most frequently used with the brasses is Tobin bronze. This melts at a slightly lower temperature than

the brasses, and alloys with the base metal avidly when the latter is at a good brazing heat.

In designing welded connections in the brasses or nickel silvers, provision should be made for holding the metal and backing it up during the welding operation. This is necessary because the metal is, unlike steel, very fluid at fusion-welding temperatures and more time can be lost in trying to patch up a hole that has melted through than it takes to do the job correctly in the first place. In some cases, as in the manufacture of metal doors, steel stiffeners can be made to serve as back-ups for the brass or nickel-silver sheet.

When the copper-alloy sheet is less than say 0.050 in. thick, fusion welding becomes difficult and silver brazing or soft solder must be resorted to. Frequently, the mechanical engineer has to design protection for bearings and moving parts which operate in an atmosphere of dust and grit. In one instance, a large bellows sheath was made up of thin, red-brass sheet, silver-brazed into cylindrical form and then corrugated for flexibility.

Resistance Spot and Seam Welding

With such alloys as common brass, nickel silver, the cupronickels and the Everdur-type alloys, satisfactory resistance spot and seam welding can be used (11). Copper sheet and commercial bronze sheet are very difficult to resistance seam-weld on account of their high thermal and electrical conductivities.

Pressure-tight welds by the resistance method in common brass are rather difficult to make although good mechanical strength is obtainable. Hence, in the manufacture of sheet-metal evaporators for small refrigerators, where a high-strength, permanent-seal type of weld is required, a silicon-copper alloy similar in its properties to Everdur is used. Such alloys possess an excellent combination of drawability, strength, resistance-weldability, and corrosion resistance.

Everdur Pressure Vessels

Perhaps the best example of fusion welding of copper alloys is the manufacture of Everdur pressure vessels (12, 13, 14). Some of these have been made by riveting with Everdur rivets but fusion-welding methods have proved most satisfactory. All sizes from resistance seam-welded kerosene containers of 1 gal capacity to fusion-welded vessels of 18,000 gal capacity are being made. The combination of high ultimate strengths of 55,000 lb per sq in. or better, corrosion resistance and general workability, including the property of weldability, make such vessels economical to manufacture and market. They are satisfactory from the standpoint of long trouble-free service.

For these reasons, many thousands of range boilers for small households and larger hot-water storage tanks for laundries, apartments, and hotels have been made and put into service. The first of these was made in 1927 or 1928 and is still in use.

Everdur may be oxyacetylene fusion-welded in all thicknesses from 0.025-in. sheet to an inch or more plate. Commercially, it is carbon-arc-welded in all thicknesses from 0.040-in. sheet to any desired thickness, and metallic-arc-welded in any thickness above $^1/_{16}$ in. Economical resistance welding is usually confined to sheet 0.080 in. or less in thickness. Satisfactory commercial, gas-tight resistance welds have been made in sheet as thin as 0.010 in.

Copper Alloys in Hydraulic Equipment

The Everdur type of copper alloy in the relatively short time it has been commercially available has found many useful applications in the valves, guides, gates, gratings, and other equipment for water supply and sewage-disposal projects.

Where the piece of equipment is made up of several parts, fusion welding with the torch or arc and Everdur welding rod has been found to be a satisfactory method of developing the required structural strength and water tightness. In several instances, Everdur anchor bars were welded to the back of the Everdur gate seats which were then grouted into the concrete dam.

A somewhat similar use was made of extruded naval-brass angles in a large hydraulic control project. In this case, the heat necessary for fusion welding was thought objectionable. Therefore, the Tobin-bronze anchor bars were threaded into nuts which previously had been silver-brazed to the back of the angles.

Aluminum Bronze, and Beryllium Copper

It is probable that aluminum bronze and beryllium copper present the most difficult problem in copper-alloy welding. This is due to the very refractory film of aluminum oxide or beryllium oxide formed on the surface of the alloy. Even though the alloy may have a mirror-bright, polished surface, the film of oxide is present in sufficient continuity to render soldering, brazing, and gas welding difficult. This is particularly true since there is no available flux that is especially efficacious in dissolving these oxides. The blanketing action of a flux can be used to advantage in soft soldering or silver soldering of these alloys. If the flux has a sufficiently low melting point and gives proper coverage, it will prevent the formation of the surface film in the heating-up stage and the soft solder or silver solder will flow to the base metal with no serious interference.

In fusion welding, the carbon arc offers a rapid and economical method of joining these alloys. The fusion of the welding rod and base metal with subsequent solidification takes place in such a brief interval that the film of oxide offers little interference and welds of high strength and soundness can be obtained. A flux is quite unnecessary although a thin wash of brazing flux to which potassium fluoride and sodium carbonate has been added will help in the flowing of the metal and minimize the oxide on the surface of the base metal.

As for applications, aluminum bronze would be used in situations where (*a*) a good forging bronze is desired, (*b*) hot hardness and nonscaling qualities are required and (*c*) where corrosion resistance is important. For (*a*) the 5 per cent aluminum bronze is commonly used. It works not unlike soft iron under the blacksmith's hammer, enabling the artisan to work out beautiful bronze shapes. It cannot, however, be hammer-welded but may be brazed with Tobin bronze or fusion-welded with the carbon arc. The high hot hardness indicated in (*b*) seems to be in conflict with the softness indicated by good forgability. However, both conditions are true. Aluminum bronze, and in particular the 8 to 10 per cent aluminum bronze, does retain its hardness at a higher temperature than most copper alloys, the thin invisible film of aluminum oxide preventing scaling. At still higher temperatures, it softens abruptly with an increase in ductility, thus making it easy to hot work.

Beryllium copper would find its economic use in places where its combination of high strength, resilience, hardness, wear resistance and corrosion resistance are needed. Numerous small parts in electrical apparatus and textile machinery can be made to advantage from this metal. Where joining is necessary, resistance spot welding or silver soldering will usually be found convenient, but in heavier parts carbon-arc welding will be the best method of joining (15).

CUPRONICKELS

Users of corrosion-resisting metals find that for certain purposes the copper-nickel alloys, carrying 10 to 30 per cent nickel and the balance copper, give satisfactory service. Used as steam-condenser tubes, where the cooling water is sea water or the brack-

ish waters of the rivers and harbors, the 70:30 cupronickel has given excellent service. Usually, these tubes are expanded into the relatively thick Muntz-metal header and welding is not required. However, such tubes are also used for salt-water lines aboard ships and in similar plumbing installations. Not only does the cupronickel resist corrosion better than most metals, but the high copper content is toxic to marine growth so that the tubes remain cleaner. For such purposes, silver-brazed connections to cast-bronze or white-metal fittings are found satisfactory. However, where fusion welds are desired, a welding rod which works very well with the oxyacetylene torch in any position has been developed (16).

2 JOINING OF FERROUS METALS WITH COPPER ALLOYS

Several instances of the joining of copper alloys to steel have already been mentioned. Many such illustrations can be cited. However, only a few of the more important can be mentioned in this paper.

Copper-Brazed Steel

One of the noteworthy economies, which has made mass production possible with its enormous duplication of automobiles, sewing machines, typewriters, adding machines, cash registers, radios, bookkeeping machines, teletype senders and recorders, refrigerators, and washing machines, has been the use of relatively thin sheet-steel parts copper-brazed together (17, 18, 19). In such manufacture, there are numerous assemblies which the maker desires to bond parts together permanently. Perhaps soft solder isn't strong enough, the metal is too thin to provide a good bearing against a rivet, spot welding not feasible and screws may work loose. For such a purpose "copper brazing" is the answer.

Credited originally to A. C. Hyde of Wolverhampton, England, the copper-brazing process (20) consists briefly of the heating of the steel to approximately 2100 F in an atmosphere of hydrogen with sufficient copper to braze the joint. At 2100 F, the hydrogen acts as an agent to reduce all iron and copper oxides allowing the copper, which melts at 1980 F, to surface-alloy with the steel and penetrate and thoroughly bond the tightest joint. Tensile strengths of 45,000 to 55,000 lb per sq in. and shear strengths of 30,000 to 40,000 lb per sq in. are obtained. In addition to light machine parts, resilient golf-club shafts made from spirally wound steel strip, tennis rackets, and an enormous footage of small sizes of steel tubing are all copper-brazed.

The copper-brazed assemblies are in the annealed, stress-free condition as they come from the brazing furnace. They may be heat-treated, carburized or vitreously enameled after copper brazing without disturbing the bond.

Although the process (17) has been known for a score of years, it has been only within the past six years that electric furnaces with atmospheres sufficiently rich in hydrogen have been made available commercially. By suitable conditions of temperature and pressure with admixtures of air or steam, natural gas, city gas and dry ammonia can be broken down, resulting in a high percentage of free hydrogen.

Spelter Brazing and Silver Brazing

A much older process than the copper-hydrogen brazing is that of spelter or silver brazing. Spelter brazing alloy is usually 50 per cent copper, 50 per cent zinc although additions of nickel may be made to whiten it, or additions of tin to harden it and reduce the melting point. Silver brazing alloys or silver solders carry varying proportions of copper and zinc with silver from 10 to 80 per cent. The spelter solders melt between 1500 and 1550 F, while the silver solders melt at temperatures from 1300 to 1510 F.

The spelter-brazing or silver-brazing processes differ from copper brazing chiefly in that a flux is invariably used and the furnace does not necessarily have a controlled atmosphere. The furnace temperature need not exceed 1750 F, and for the silver solders, it may be lower.

Tensile strengths of 50,000 lb per sq in. are obtained readily in spelter- or silver-brazed steel connections with correspondingly good shear strengths. Normally, the steel parts cannot be heat-treated, carburized, or vitreously enameled after the brazing process. Excess flux must be removed before plating, cold enameling or other finishing processes.

The brazing-in of the heads of small steel tanks for high-pressure air is a variation of the process in which the spelter solder is first melted in a crucible and the tank fluxed, preheated, and dipped into the molten brazing solder. The spelter solder makes a strong, permanent, gas-tight seal.

Bronze Welding

Bronze welding is a process of great industrial importance. Several million pounds of bronze welding rods are consumed annually. While the bronze is deposited on steel or cast iron with a surface-alloying effect similar to that obtained in copper or spelter brazing, the process is allied very closely to fusion-welding methods. It has the economic advantage of fusion welding of cast iron, steel or copper in that the welding temperature is much lower. It probably possesses the widest range of applications for the joining of dissimilar metals.

The bronze welding rods referred to are the beta brasses carrying 38 to 45 per cent zinc with additions of tin, iron, or manganese up to 1 per cent. Tobin bronze and manganese bronze are two examples of the welding bronzes. Owing to the high zinc content, which volatilizes freely at temperatures above 1700 F, the arc should not be used for the welding heat.

Tensile strengths in the bronze weld metal itself vary from 40,000 to 55,000 lb per sq in. The strength of the bond to cast iron and steel varies considerably with the preparation and size of deposit. On properly bronze-welded steel, the bond strength should be 40,000 to 50,000 lb per sq in. On cast iron, it will be less owing to the interference from the graphite or temper carbon.

The process is applicable to any thickness of cast iron (21), malleable cast iron or steel, cast or rolled, bare or galvanized. It is being used with equal facility on 0.020-in. thick steel or copper-alloy sheet or the 8 × 8-in. section of a cast-steel locomotive frame. As a manufacturing process, it is used on a great variety of products. As a repair tool, it is practically indispensable for the maintenance of any industrial, agricultural, or transportation machinery.

Bronze welding affords an easy method of joining copper, brass, and galvanized-iron pipe for interior plumbing and cast-iron pipe for underground service.

One special application of torch or carbon-arc brazing is the joining of thin sheet steel with Everdur welding rod. Thus, in the manufacture of metal furniture, Everdur may be applied with the oxyacetylene torch or the carbon arc without fusion of the thin sheet steel. No flux is necessary and there is very little distortion.

Another special application of bronze welding is the joining of stranded-copper rail bonds to the head of the steel rails. This may be done by the use of the oxyacetylene torch and a high-strength yellow bronze such as Tobin bronze but more frequently it is done with the carbon arc and a high-conductivity copper-alloy welding rod (6). Owing to the fluidity of the weld metal, a carbon mold is usually clamped to the head of the rail with the bond.

Surfacing With Copper Alloys

Welding methods offer the mechanical engineer a means of coat-

ing new surfaces with a wear-resisting copper alloy or restoring worn surfaces to their original condition. The excellent wear resistance of manganese bronze and phosphor bronze against cast iron and steel are well known. Less well known are Everdur and beryllium copper and wear-resisting alloys.

The manganese bronze should be deposited with the oxyacetylene torch while phosphor bronze and Everdur are applied to cast iron or steel by the metallic arc or the carbon arc. Beryllium copper can be applied only by the carbon arc.

In the endeavor to increase the life of a locomotive between shoppings, the railroads find it economical to coat the pistons, pedestals, and driver boxes with bronze. In some cases, the crosshead shoes are also bronze-coated.

Manganese bronze applied with the oxyacetylene torch has been used on the pistons for the most part. These pistons in the older locomotives were made of cast iron in which case this was the optimum process. More recently, cast steel is being used for the piston heads in which case the designer has the choice of specifying manganese- or phosphor-bronze surfacing by oxyacetylene torch, or phosphor bronze applied by the carbon or metallic arc.

Phosphor-bronze plates welded into place have in some cases been used as liners for the locomotive pedestals to take the thrust and wear from the front and rear faces of the driver boxes. The outside faces of the cast-steel driver boxes are usually coated with Everdur or phosphor bronze applied by the welding arc or manganese bronze by the welding torch.

One industrial application of hard surfacing is the use of hard, wear-resisting beryllium copper as a face for the jaws of a resistance or flash butt-welding machine. The beryllium copper face can be silver-brazed onto the water-cooled, high-conductivity copper backer, good results obtained in some cases. However, if the beryllium copper is welded directly to the high-conductivity copper electrode, a maximum in heat and electrical conductivity is obtained. This desirable combination can be obtained by the use of an extremely hot carbon arc which melts the beryllium copper and heats the heavy copper base metal to the temperature at which the weld metal will surface-alloy thereto.

CONCLUSION

In general, metals will find their applications of greatest economic usefulness as surely as water will find its level. Owing, however, to the slower operation of the economic laws and the still slower human understanding of these laws, metals are often misplaced. Again, in the case of a long-life metal as copper, the monetary yard stick changes faster than the metal itself, making it difficult to estimate service costs on a per-annum basis. In many cases trouble-free service means more than the initial cost. Hence, the copper alloys, though higher priced may be very properly specified.

In the foregoing illustrations, the author has indicated some of the applications in which welded copper alloys are being successfully used. Many more cases could be cited. In general, it is believed that there is one best metal for a given purpose. This thought is creeping into the consciousness of the metal manufacturer whose thoughts several decades back were given solely to tonnage figures. More and more metals are being prepared especially for a given use. More and more the mechanical engineer and metallurgist alike realize that the proper use of a metal is an important feature of the design and that welded connections often offer the best solution of a vexing design problem.

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The Electric Welding of Monel and Nickel

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The authors discuss the metallic-arc and carbon-arc welding of monel and nickel as well as the welding of these two products to steel plate. They also give the chemical and physical properties of the two metals and the chemical and physical properties of deposited weld metal.

Sheet and Plate of Monel and Nickel

About six years ago a rather complete change was made in the melting and refining practice of monel and nickel. Subsequent to that time the hot ductility of both materials has been much improved. With material of current production it is possible to make hot-ductility, 180-deg free bend tests on all heats at 100-F increments through a complete temperature range of 1200 to 2200 F. Material of earlier production was hot short between 1200 and 1600 F. The improvement in hot ductility as measured by reduction in area was quite marked, there being a rise from about 25 per cent for earlier production to 50 per cent for current production in tests made at 1800 F. Because of the improved production methods, hot working of monel and nickel in the forming of heavy equipment no longer requires extremely accurate temperature control, and the metals can now be worked in a temperature range considered dangerous a few years ago. Another important advantage of monel and nickel produced by current methods is freedom from cracking during welding. The nominal compositions for standard wrought monel and nickel products are given in Table 1. The physical properties of monel and nickel are given in Tables 2 and 3, respectively.

Electrodes

About five years ago the manufacture of flux-coated electrodes of monel and nickel was begun. At the same time the study of arc welding these two metals was broadened. Regular cold-drawn monel wire was used for many years for the core wire of electrodes. In the light of present practice, results were only fair, although a few years ago they were considered quite satisfactory. It was found that where a light wash coating of a mineral flux gave only fair arcing characteristics, an improvement in arcing characteristics was obtained where a slagging type of coating was used. The thickness of coating was multiplied several times, but since the thicker coatings were applied by hand dipping, the problem of concentricity became paramount. Attempts at applying several thin coatings to build up a heavier total thickness were mildly satisfactory. It was found also that with these plain dip coatings, a slight amount of porosity still persisted in the deposited weld.

The usefulness of aluminum as a deoxidizer in nickel alloys was known, but advantage had not been taken of this in welding. Finally it was decided that the use of aluminum in the form of fine wire wrapped in a spiral around the 18-in. length of bare electrode, would serve a twofold purpose: First, it would aid in deoxidation, and it was hoped, eliminate porosity, and second, the spiral would serve as an excellent support for the flux coating applied by dipping. This procedure worked out very satisfactorily with both monel and nickel and was used for several years. The more recent developments of coating by extrusion

TABLE 1 NOMINAL CHEMICAL COMPOSITION OF MONEL AND NICKEL

Element	Monel, per cent	Nickel, per cent
Nickel	68.00	99.40
Copper	29.00	0.10
Iron	1.50	0.15
Manganese	1.10	0.15
Silicon	0.10	0.10
Carbon	0.15	0.10

TABLE 2 PHYSICAL PROPERTIES OF MONEL

	Tensile strength, lb per sq in.	Yield point, lb per sq in.	Elong. in 2 in., per cent
Sheet and strip:			
Cold-rolled			
Annealed	65000– 80000	25000– 35000	40–25
Full hard	100000–120000	90000–110000	8–2
Plate:			
Hot-rolled			
As rolled	80000–110000	40000– 90000	55–27.5
Annealed[a]	70000– 85000	25000– 45000	
Tubing:			
Cold-drawn			
Annealed	65000– 80000	25000– 35000	35–25
As drawn	90000–105000	60000– 75000	20–15
Rod and bar:			
Cold-drawn			
Annealed	70000– 85000	25000– 35000	50–35
As drawn	85000–125000	60000– 95000	35–15
Hot-rolled	80000– 95000	40000– 65000	45–30
Wire:			
Cold-drawn			
Annealed	70000– 85000	25000– 35000	45–25
Spring	140000[b]	....	8
	160000[c]	....	2

[a] Annealed between 1700 and 1750 F for 5 min.
[b] Brown and Sharpe gage of 0 to 2.
[c] Brown and Sharpe gage of 15 to 19.

TABLE 3 PHYSICAL PROPERTIES OF NICKEL

	Tensile strength, lb per sq in.	Yield point, lb per sq in. (0.5% set)	Elong. in 2 in., per cent
Sheet and strip:			
Cold-rolled			
Annealed	60000– 75000	15000– 25000	45–35
Full hard	100000–115000	90000–105000	10–2
Plate:			
Hot-rolled			
As rolled	70000–100000	30000– 75000	60–30
Annealed[a]	65000– 80000	20000– 27000	65–55
Tubing:			
Cold-drawn			
Annealed	60000– 75000	15000– 25000	45–35
As drawn	80000– 95000	50000– 60000	25–15
Rod and bar:			
Cold-drawn			
Annealed	65000– 85000	20000– 30000	50–35
As drawn	80000–115000	60000– 90000	35–15
Hot-rolled	70000– 85000	20000– 30000	45–35
Wire:			
Cold-drawn			
Annealed	80000– 95000	....	45–35
Spring	140000–175000	....	10–5

[a] Annealed between 1650 and 1700 F for 5 min.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

were applied, with the result that the extruded monel electrode is commercially available now, whereas, the hand-dipping practice is still followed in the case of nickel.

Inasmuch as coating by extrusion does not conveniently permit a metallic spiral aluminum winding, and as the presence of aluminum appeared desirable in or near the molten pool of weld metal, experimental activity was directed along the lines of introducing this element in one of two forms, either in the coating or in the core rod. The latter was found more convenient and a small amount of aluminum in monel wire for welding has been found to be entirely satisfactory from the standpoint of welding characteristics, physical properties, and soundness in the deposited metal.

Up to the present time only the monel electrodes have been

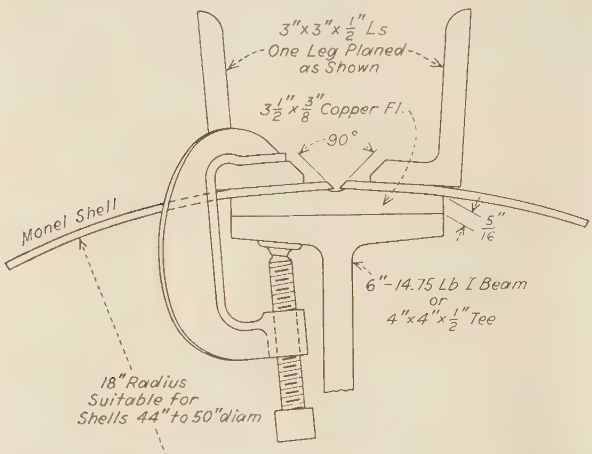

FIG. 1 JIG FOR WELDING SHORT LENGTHS OF THIN-GAGE MONEL AND NICKEL

coated by extrusion for commercial use. Following growth of knowledge and experience with the exacting requirements of extrusion coatings, as to pressures, speeds of extrusion, centering and, of great importance, proper drying, the monel electrodes now produced are of highly satisfactory quality. The criteria are, good arcing characteristics and ability to deposit strong, sound, ductile metal consistently under a variety of operating and shop conditions.

The presence of aluminum in quantities greater than those normally occurring in regular monel welding rod or sheet are to be found in that composition of monel known as type *K*, and available in various rolling-mill forms. This material may be hardened by heat-treatment, as well as by cold working, and the effects are additive. Regular monel is not amenable to similar hardening by heat-treatment. This heat-treatment hardening of type-*K* monel is accomplished by a form of precipitation hardening as with dural.

The use of a slightly alloyed monel rod is developing strengths of 70,000 to 80,000 lb per sq in. in single-bead and multiple-bead butt joints, metallic-arc welded. This is the range of tensile strengths obtained in plate material. It is possible to get still higher values using a core wire of type-*K* monel. A reference to the higher strengths available with these rods will be given later in the paper under the heading: "Quality of deposited metal."

Higher strengths in the weld metal than in the parent material would hardly be useful, since it is the strength and ductility of the completely welded joint which is of interest to designers and not merely strengths of plate or weld metal separately.

By its very nature, molten nickel has a great affinity for gases,

TABLE 4 PHYSICAL CONSTANTS OF MONEL AND NICKEL

	Monel	Nickel
Density, rolled, g per cc	8.9	8.9
Weight, rolled, lb per cu in	0.323	0.323
Melting point, F	2460 (1350 C)	2640 (1450 C)
Coefficients of expansion between 70 and 1100 F, in. per in. per deg F	0.0000091	0.0000085
Modulus of elasticity in tension, lb per sq in.	25000000 26000000	30000000
Modulus of elasticity in torsion, lb per sq in.	9000000 9500000	10000000

and when it freezes or solidifies, these gases are thrown off to form gas pockets or porosity, certainly an undesirable characteristic in welding. It was this condition which necessitated an investigation, made about ten years ago, with the object of making small additions to essentially pure nickel. These additions were important alloying elements which might control the porosity. The outcome was a composition of nickel welding wire which had controlled amounts of titanium and magnesium and which measurably reduced the gaseous condition. This special nickel welding wire was adopted and is still standard for gas and electric welding of pure nickel. With gas welding, no porosity is evident, but with metallic-arc welding of nickel it does occur occasionally.

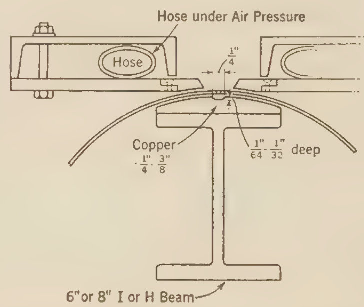

FIG. 2 JIG FOR WELDING LONG LENGTHS OF THIN-GAGE MONEL AND NICKEL

Current investigations are showing material progress in the elimination of this porosity.

METALLIC-ARC WELDING OF MONEL AND NICKEL

Monel electrodes and nickel electrodes both carrying relatively thick flux coatings require reversed polarity as do most of the high-strength steel electrodes. While no attempt has been made to determine the reason why this polarity is productive of more satisfactory metallic-arc welds, suffice it to say that reversed polarity by experience is the more suitable.

In the metallic-arc welding of sheets of light to medium gage between 0.037 in. and 0.125 in. thick, it has been found desirable to use a means of clamping or jigging to restrain buckling. While this recommendation has been made in connection with monel and nickel welding for many years, industry has sometimes referred to it as an apparent shortcoming of these two metals to so prepare joints prior to welding. However, it is just as desirable in the welding of light-gage sheet steel. The coefficients of expansion are cited in Table 4 and it should be noted that those for monel and nickel are practically identical with the coefficient of expansion for carbon steel which is 0.0000085 in. per in. per deg F for the temperature range of 100 to 1300 F.

Using identical gages in either monel, nickel, or steel, and identical setups and welding processes, will give essentially the same results as to degree of buckling. With progress in improving the properties of steel weld metal, there has come also an improvement in appearance, that is, freedom from buckling of steel-fabricated pieces, which is due in no small measure to wider use of jigs and clamping devices. Of course, sequence in welding

also has been an important factor. No longer can the suggested use of simple jigs be considered a liability in a monel or nickel fabrication, but a case of necessity where close limits must be held and where buckling or warping of surfaces will cause rejection.

For welding longitudinal butt seams, simple jigs such as shown in Figs. 1 and 2 are frequently used. Particularly is this the case where longitudinal joints are to be made in cylinders where proper alignment of sheet edges is of primary importance and where perfect roundness is necessary.

The jigs suggested for use in butt joints are of greatest value in sheet thicknesses up to about $^{1}/_{8}$ in. or slightly thicker. It can be seen that a jig serves two purposes: First, to hold the sheets in line during welding, thus making welding easier and obtaining a joint freer from buckling, and second, since a grooved backing up strip is provided this shallow groove acts as a mold for the weld metal and permits more uniform penetration, the latter being an important factor where the penetration side of a single bead joint must be ground and polished flush with the sheet surface. Either or in fact both sides of the welded joint can be ground and polished to a high finish.

On fillet welds, where it is convenient, and certainly on thin materials, it is desirable that an angle be used to back up a joint. It is not necessary to back up corner welds in material greater than $^{1}/_{8}$ in. thick.

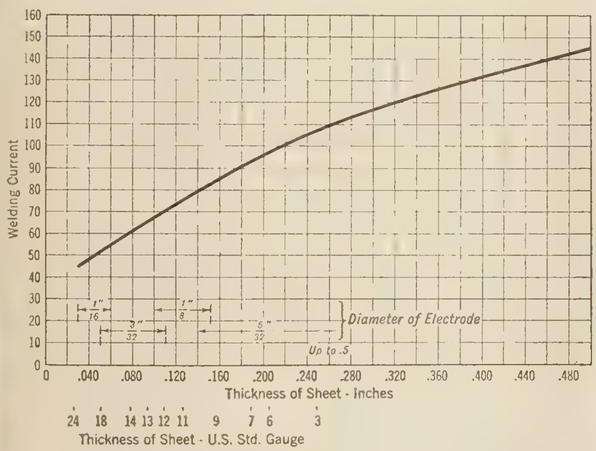

Fig. 3 Relation Between Sheet Thicknesses and Current Required for the Metallic-Arc Welding of Monel and Nickel

(The approximate welding currents for metallic-arc welding given are useful in making a setting on the welding machine, and should be used as a guide only. Such factors as gage of sheet, electrode diameter, backing material, and jigging or clamping all affect the current setting.)

In many cases, it is uneconomical or inconvenient to use jigs or clamps when welding medium or light gages. It would be unwise to burden any cost sheet with the cost of jigs, the usefulness of which will be extremely limited. However, once jigs of the type indicated in Figs. 1 and 2 are available in any shop, its wide applicability will be readily appreciated even on the garden variety of sheet-steel work.

In the metallic-arc welding of monel and nickel, there are several factors which affect the heat or amperage required for a particular job. Some of these are the diameter of electrode, the gage of sheet, the speed of welding and, coupled with the latter, the degree of penetration, type of backing up, whether copper, steel, or unsupported, and if a jig is used, whether the metal is tightly clamped to it. It is because of these many factors that accurate heats are not readily available. The curve in Fig. 3 gives the approximate relation of sheet thicknesses to current required for metallic-arc welding monel and nickel.

For the lighter gages, it has been found that beads made without weaving are entirely satisfactory. There is no definite dividing line where straight-line welding should be stopped and weaving begun. The $^{5}/_{32}$-in. thick sheet for butt or lap joints can be used as a convenient dividing gage above which a slight weave may be found desirable. For fillet welds the approximate dividing line is $^{1}/_{8}$ in., but a weave is definitely preferred for sheets above $^{5}/_{32}$ in. thick.

The angle of the electrode in making butt welds should be from the vertical position to about 30 deg from the vertical. The electrode should be about normal to the molten pool of weld metal. This electrode position permits a more uniform flux coverage of hot or molten metal, and is to be preferred to the practice sometimes found where the electrode position is such as to push molten slag ahead of the welding pool, in the root of the V.

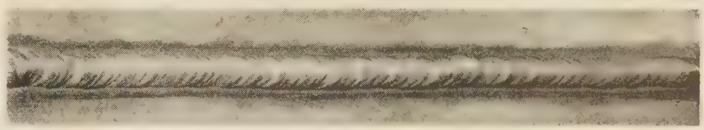

Fig. 4 A Typical Carbon-Arc Weld in Monel

In fillet welds, where metal thicknesses are the same, a 45-deg electrode position, bisecting the fillet, is good. Of course, where dissimilar thicknesses are to be joined, then the electrode should be so positioned that the arc plays on the heavier of the two pieces.

For horizontal, vertical, or overhead welding, practices regularly followed in steel welding are followed. Wherever possible, it is naturally desirable that positioning of work be such that welding is in the flat position or nearly so. Certainly better control is obtained, and unquestionably better weld metal is the result. Position welding can be done, and is being done, on jobs which cannot be moved, and the end results, the physical properties of weld metal, and the appearance of the deposited bead, are entirely satisfactory.

Carbon-Arc Welding of Monel and Nickel

While the carbon-arc process does not receive much mention in the literature, it nevertheless fills a certain need which is difficult to explain but is definitely known to exist. Specifically, carbon-arc welding is frequently applied to monel and nickel sheet in the intermediate range of gages of 0.037 in., 0.050 in., and 0.062 in., and oftentimes heavier. Because of the somewhat slower speed of carbon-arc welding occasioned by the slightly lower rate of heat input, it is possible to control quite accurately the height of bead and degree of penetration. Particularly is this the case where a very small, narrow bead is to be deposited.

Some rather large fume ducts were to be made of 0.050-in. monel sheet, and one type of joint used for field erection was a lap joint in which one sheet was offset, or jogged, slightly. The lap joint was tacked on the outside, then the entire interior was carbon-arc-welded to seal the joint and make it airtight. All carbon-arc welding was done in position with more work being done in the vertical and overhead positions than in the flat position. A lightly fluxed monel filler rod for carbon-arc welding was used in all cases, with carbon pencils of $^{5}/_{32}$ in., $^{3}/_{16}$ in., and $^{1}/_{4}$ in. diameter, pointed like a pencil for at least 2 or 3 in. The type of seam obtainable by taking advantage of carbon-arc welding on monel and nickel is shown in Fig. 4. The uniform ripple surface can be obtained readily with steady hands and experience.

Dry-cleaning equipment of the type shown in Fig. 5 is manufactured of monel sheet of various gages, but largely of 0.050 in. and 0.062 in. thicknesses. Butt and corner welds are all made using a $^5/_{32}$-in. diameter carbon pencil and a monel carbon-arc welding wire. Here again, uniform appearance and pressure tightness without the necessity of grinding and polishing are required.

Beside being useful for medium-gage welding, carbon-arc welding is equally useful for joining small-diameter tubes into tube sheets. Tubes are left projecting $^1/_8$ in. to $^1/_4$ in. above the surface of the tube sheet and then either melted down with the

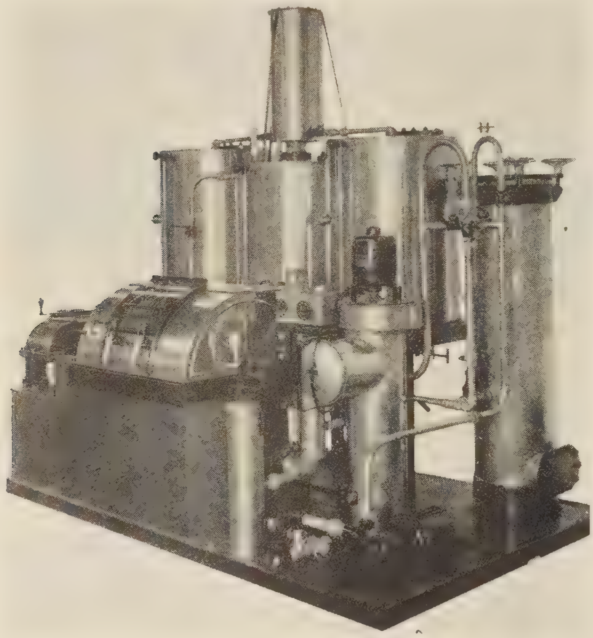

Fig. 5 Dry-Cleaning Unit Fabricated by Welding Various Gages of Monel

carbon arc or a neat fillet deposited using the lightly fluxed carbon-arc filler rod of appropriate metal composition.

The ductility and physical properties of carbon-arc welds while satisfactory, are not as high as those obtainable with metallic-arc welds in monel or nickel. Where complete 180-deg bends can be made on metallic-arc welds, bending transverse to the direction of welding, bends of the order of 160-deg can regularly be made in the case of carbon-arc welds.

Where the small-diameter carbon or graphite pencils are used, close control of the pool of molten weld metal is always obtained if care is taken to retain a sharp tapered point on the carbon. If the taper is 2 or 3 in. long, and the carbon is gripped in the electrode holder about 4 in. from the tip of the carbon, a fairly uniform wasting away of the carbon will maintain the point, and prevent it from becoming blunt. Just as soon as the end becomes blunt, the arc begins to wander, and the control of the pool is slightly reduced.

The polarity for carbon-arc welding is straight polarity, that is, the work is positive and the electrode is negative. A steady, uniform and quiet arc is obtainable without undue wasting of the carbon or graphite electrode. If reverse polarity is inadvertently attempted, the difference in polarity will immediately be evident in the hissing, spluttering sound, as well as in the fact that the molten pool will be difficult to control.

The length of the arc is maintained between $^1/_{16}$ and $^1/_8$ in., a relatively short arc when compared to the long one required for copper carbon-arc welding. However, the short arc for monel and nickel can be varied slightly, that is, the arc length can be increased or decreased a little to obtain a quiet, smooth arc.

The position of the carbon electrode, and monel or nickel filler rod with respect to the molten pool of weld metal is exactly the same as when acetylene welding in the forehand position, that is, the torch pointing in the direction of the unwelded seam. The carbon electrode is held at an angle between 30 and 60 deg. The filler rod is held at the same angle. It is helpful, when starting to weld on cold metal, to use the following procedure. The point of the carbon pencil is placed in contact with the metal at the beginning of the joint to be welded, but an arc is not struck; instead, the carbon is held in contact with the metal to be welded for about 10 sec. This short-time contact serves not only to preheat the work slightly, but also brings the carbon to a red heat quickly and, accordingly, uniform heat conditions exist and the weld can then be started immediately. The curve in Fig. 6 gives the approximate relation of current to sheet and plate metal thickness.

Carbon-arc welding is useful in certain instances where oxy-acetylene and metallic-arc welding cannot always meet exacting requirements of uniformity of appearance and penetration.

Welding Dissimilar Metals

It often becomes necessary to join dissimilar metals, such as monel to steel, and nickel to steel. A procedure, which has been

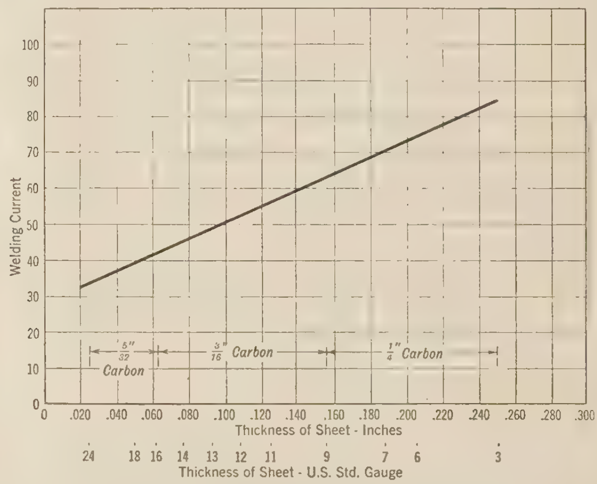

Fig. 6 Relation Between Sheet Thicknesses and Current Required for the Carbon-Arc Welding of Monel and Nickel
(This curve is for welding without a backing. Where copper or other metal backing is used, it is necessary to add 5 or 10 amp to the values shown.)

used successfully for a number of years on $^3/_{16}$ in. and heavier metal, in joining nickel to steel, is of interest. In the electric welding of a butt joint, the edges are beveled to 35 deg, then an overlay, or thin weld, of nickel is deposited on the scarfed edge of steel. This overlay or weld of nickel, as shown in Fig. 7, really presents a nickel surface, which can then easily be welded to the nickel plate with one or more beads. Again in the case of welding nickel pipe connections to the steel side of a nickel-clad steel vessel, it is desirable to punch the hole through the plate, cover the periphery of the hole with an overlay of nickel weld metal, insert the tube in place, then lay one or more nickel metallic-arc welds to complete the joint on the outside fillet.

If, instead of the procedure just indicated, a single-fillet weld were made between the nickel tube and steel plate to make the joint in a single pass, it would probably result in a checking in the

TABLE 5 TEST RESULTS OF NICKEL-TO-STEEL BUTT JOINTS IN $^{3}/_{16}$-IN. PLATE METALLIC-ARC WELDED

Type of tensile specimen	Failure	Tensile strength, lb per sq in.
Reduced section with weld reinforcements removed	In weld	57200
Full-section with weld reinforcements retained	In steel	57900

NOTE: In a free-bend test, the outermost fiber in the weld elongated 9.6 per cent before failure occurred.

weld. Regardless of whether a nickel or steel electrode is used, the single-bead fillet weld between a nickel and steel surface is not foolproof.

Where the correct procedure is followed on butt joints for veeing and for overlaying steel with nickel and then welding with nickel, the results given in Table 5 have been obtained.

When welding monel to steel it has not been found necessary to use the procedure of overlaying the steel surface as has been found desirable for welding nickel to steel.

In the joining of $^{1}/_{4}$-in. monel to steel, a single V weld was made with a 70-deg included angle. The butt joint can be welded with a monel electrode, with good results. The macrograph and the micrographs shown in Fig. 8 naturally reveal quite a difference in appearance between welds and the base metals. Tensile tests of two monel-to-steel welded specimens with weld reinforcement removed gave tensile strengths of 57,750 and 61,500 lb per sq in., respectively. Both specimens failed in the steel.

These strengths, since they are normal values for the steel-plate material, are satisfactory for the welding of dissimilar metals, and will suffice where this type of welding must be done. However, the composition of the weld metal is quite unknown, and empirical results are unfortunately the only source yielding sound information.

Quality and Physical Properties of Deposited Metal

With a weld on thin sheet, say 0.062 in., it is possible to obtain the strength of the joint, but to attempt to measure the yield point is absurd because actually any value obtained is not a true value for either the sheet or the weld. Similarly, if the break is in the sheet, the ductility of sheet material is determined. With thicker metal, say $^{1}/_{8}$ in. or $^{1}/_{2}$ in., the tensile values are easily obtainable again, but yield strengths of welded joints are still of little value, except when all-weld-metal specimens are used.

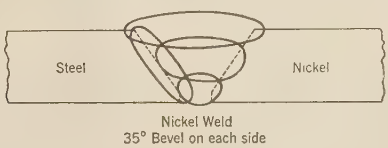

FIG. 7 METHOD OF WELDING NICKEL PLATE TO STEEL PLATE

In lieu of suitable ductility values being obtained in a tensile test, recourse has been taken to either a free- or jig-bend test. This type of test has proved quite useful and consists merely of taking a $1^{1}/_{2}$-in. wide section of weld, in the approximate form of a tensile specimen, and bending the piece so that the stresses all fall in the weld, which is parallel with the bend.

Table 6 gives values of welds recently made as a part of welder approval tests. The weld test plates were allowed to cool to 212 F before the next bead was deposited. Exographs of all the plates in connection with this test were sound and dense.

Test plates of $^{1}/_{8}$-in. monel plate made up for welder qualification tests are sufficiently large to make available specimens for standard, reduced-section short-gage and long-gage tensile coupons, and two free-bend coupons. In the tensile specimens, the weld reinforcement is removed and the weld is normal to the applied tension.

Bend specimens are similar to those called for in the A.S.M.E. Unfired Pressure Vessel Code, but inasmuch as the gages involved are considerably less than the steel-plate thicknesses, the welded monel bend specimens are naturally much thinner. Two types of bend are made, one with the face of the weld in tension and the other with the root or penetration side of the weld in tension. The purpose is to determine the ductility of the deposited weld metal and the quality of the bond or fusion zone. A bend is marked as "satisfactory" if it has no failure longer than $^{1}/_{16}$ in. In all cases the bend is around a $1^{1}/_{2}$-in. diameter pin.

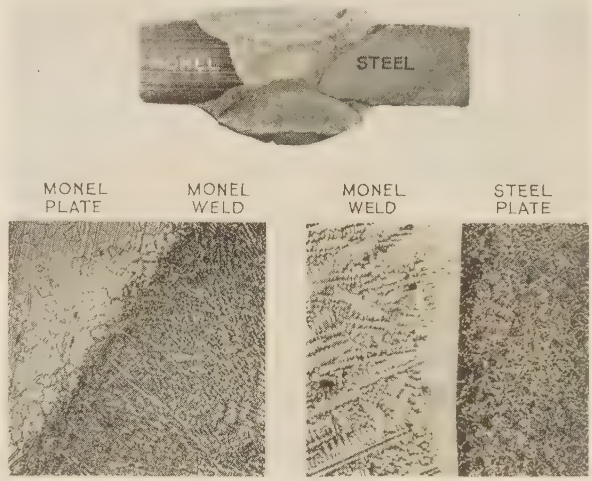

FIG. 8 PHOTOMACROGRAPH AND PHOTOMICROGRAPHS OF MONEL METAL WELDED TO STEEL PLATE

Table 7 shows results of welder qualification tests on $^{1}/_{8}$-in. monel sheet. Each weld was made by a different welder. This was all single-bead work in which all weld reinforcement was removed before testing in any way. Incidentally, in connection with this same series of tests, a number of X-rays were made of the welds. These exographs were all satisfactory.

Similar test work was carried out on $^{3}/_{8}$-in. hot-rolled monel plate. The weld test plate was set up as a single V butt joint, with a 70-deg bevel and a $^{1}/_{8}$-in. space between plate edges at the base of the V. Metal from a $^{5}/_{32}$-in. diameter monel electrode

TABLE 6 RESULTS OF WELDER APPROVAL TESTS ON $^{3}/_{8}$-IN. MONEL[a]

Specimen	Condition (machined all sides)	Dimensions of test piece, in.	Tensile strength, lb per sq in.	Jig bend 180 deg around $^{3}/_{4}$-in. radius
A-1	As welded	0.373 × 1.005	84500	....
A-2	As welded	0.380 × 1.015	79400	....
A-3	As welded	0.375 × 1.500	...	Root bend—O.K.
A-4	As welded	0.375 × 1.500	...	Face bend—O.K.
B-1	Heated[b]	0.376 × 1.015	85700	....
B-2	Heated[b]	0.376 × 1.020	90400	....
B-3	Heated[b]	0.375 × 1.500	...	Root bend—O.K.
B-4	Heated[b]	0.375 × 1.500	...	Face bend—O.K.

[a] Published through the courtesy of the Newport News Shipbuilding and Drydock Company.
[b] Heated to 750 F and held for one hour.

TABLE 7 RESULTS OF WELDER QUALIFICATION TESTS[a]

Number	Long-gage tensile test, lb per sq in.	Short-gage tensile test, lb per sq in.	Face bend	Root bend
1	81900	81600	O.K.	O.K.
2	83700	77200	O.K.	O.K.
3	79900	83000	O.K.	O.K.
4	75000	75660	O.K.	O.K.
5	74100	75000	O.K.	O.K.

[a] Published through the courtesy of the Newport News Shipbuilding and Drydock Company.

TABLE 8 CHEMICAL ANALYSIS OF MONEL WELD METAL

Element	Weld metal, per cent
Nickel plus cobalt	66.98
Copper	29.75
Iron	0.79
Silicon	0.50
Manganese	0.36
Aluminum	1.17
Sulphur	0.009

TABLE 9 PHYSICAL PROPERTIES OF $^3/_8$-IN. MONEL METALLIC-ARC WELD[a]

Test	Tensile strength, lb per sq in.	Ductility, per cent	Break
Full-section tensile specimen	78500	..	In weld
Reduced-section tensile specimen	77600	..	In weld
Free bend, face of weld in tension, 180 deg bend	...	44.4	No failure
Free bend, reverse of weld in tension, 180 deg bend	...	50.0	No failure

[a] Published through the courtesy of the research department of M. W Kellogg Company.

TABLE 10 HARDNESS SURVEY OF $^3/_8$-IN. MONEL WELD[a]

Location of test	As welded	After stress relieving at 1150 F
Center line of weld:		
Top	159	156
Center	159	165
Bottom	141	141
Junction of weld and base metal	159	154
Center of heat-affected zone	153	153
Base metal remote from weld	144	144

[a] Readings taken as Rockwell and converted to Brinell. Published through the courtesy of the research department of M. W. Kellogg Company.

was deposited at a current between 130 and 140 amp, and a load between 25 and 30 v. The piece was not allowed to cool between beads. The usual seal weld was deposited in the chipped groove on the back side of the weld. Prior to physical testing, the welded plate was stress relieved at 1150 F for the usual one hour per inch of thickness. The X-ray of the weld was free from any porosity and was characterized as "perfect, no defects." Millings of weld metal retained from machining of test specimens and later used for chemical analysis of weld metal yielded the results given in Table 8. The physical properties are shown in Table 9. Results of a hardness survey of a cross section of the weld are given in Table 10 while the photomacrograph of such a weld is shown in Fig. 9.

The $^1/_2$-in. monel plates, beveled to $37^1/_2$ deg each, were set up for a butt joint, welded with a special monel electrode containing more than the usual amount of aluminum, and the plate was allowed to cool between passes. The weld was made with four beads deposited in the V, and a single seal bead laid in on the reverse side with a $^5/_{32}$ in. electrode, all at 145 amp and 28 v. The results shown in Table 11 were obtained after stress relieving for 3 hr at 1077 F (575 C).

TABLE 11 PHYSICAL PROPERTIES OF HIGH-STRENGTH MONEL WELDS[a,b]

Weld specimen	Proportional limit, lb per sq in.	Yield strength (0.2% set), lb per sq in.	Tensile strength, lb per sq in.	Elong. in 1 in., per cent
1	33200	46000	93440	42.0
2	26000	45200	93120	41.1

[a] Values obtained after stress relieving at 1077 F for 3 hr.
[b] Published through the courtesy of the Watertown Arsenal.

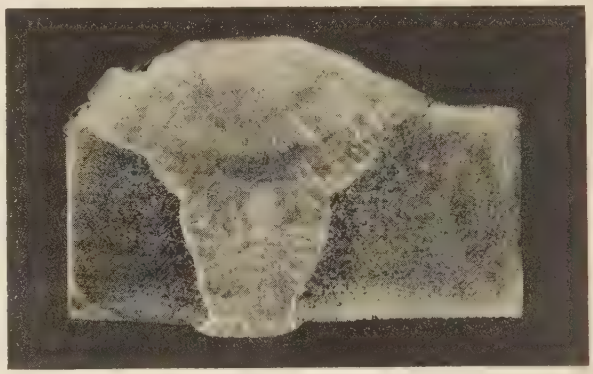

FIG. 9 MULTIPLE-BEAD METALLIC-ARC WELD IN $^3/_8$-IN. MONEL PLATES

Although by far the greater activity has been with monel, nevertheless, much nickel is being joined by fusion welding. Whenever heavy or large vessels require the use of nickel on their interior surface, solid nickel is not used, but instead, a bimetal known as nickel-clad steel. This is flange-quality steel with a thin veneer or cladding of pure nickel on one surface. As the welding of nickel-clad steel is a subject apart from this paper, and as it is already ably covered by a technical bulletin[3] now available, only brief mention of it will be made. In the case of a butt joint, the steel side is beveled with either a V or U groove and that side completely welded in, following regular steel practice. As a continuous nickel interior is desired, the nickel side of the seam is chipped out with a diamond point or round nose gouge, and then one or more beads of nickel are deposited with the metallic arc. The strength of a butt joint in nickel-clad steel plate, available in thicknesses greater than $^3/_{16}$ in., is essentially the same as a properly made weld in steel plate, and designs are based on this value.

There is much improvement in the welding of monel and nickel, and in view of the very satisfactory properties obtained, this method of joining is to be recommended for these two materials.

[3] "Methods for the Construction of Nickel-Clad Steel Plate," by F. P. Huston and T. T. Watson. Published and distributed by the International Nickel Co., New York, N. Y.

The Welding of Aluminum Alloys

By G. O. HOGLUND,[1] NEW KENSINGTON, PA.

The author discusses the use of gas, metallic-arc, and electric-resistance welding of aluminum but points out that although other forms of welding are applicable to the fabrication of aluminum products, none as yet are of commercial significance. The author includes references to the types and gages of aluminum alloys suitable for welding, the difficulties encountered in welding the alloys, the types of equipment used in the welding processes, and methods in use for the fabrication of aluminum products. He also mentions some of the industrial applications where welded-aluminum products can be used profitably.

WELDING processes for joining structures of almost all engineering materials are now well-established in industry. It follows that the growing use of aluminum and its alloys has necessitated adaptation of the common welding processes to the physical characteristics of this metal. A considerable amount of research, development, and practical work has been done in this connection, and it is the purpose of this paper to review the present status of the welding art as it is used currently in fabricating the aluminum alloys.

There are three welding processes at the present time which are of commercial significance in assembling aluminum parts. These are torch or gas welding, metallic-arc welding, and electric-resistance welding as exemplified by spot and seam or line welding. There are, in addition, a number of other welding processes which have been adapted to joining aluminum experimentally or in limited production. These include butt and flash welding, manual carbon-arc welding, atomic-hydrogen welding, automatic metallic- and carbon-arc welding, and several others. While good sound welds have been made with these processes, and some of them offer considerable promise with further development, the production possibilities of these methods have not, as yet, been apparent in practice. Further reference to the latter methods will not be covered here, though questions are invited in this connection where these processes may be of specific interest.

Torch or Gas Welding

The intelligent application of gas welding in production involves an understanding of the metallurgical characteristics of the aluminum alloys. This phase is relatively more important with gas welding than with other welding methods because the relatively high thermal conductivity and expansivity of aluminum causes a comparatively wide area to be affected by the welding heat. These factors are not difficult or complicated to understand and the effect of the welding temperature on the constituents, heat-treatment, or strain hardening are expressed as follows.

Considering first the materials suitable for gas welding, commercially pure aluminum (2S) and aluminum of higher purity are readily welded. Welds in these materials are characterized by a strength better than that of the parent material, good ductility, and excellent resistance to corrosion. Where higher strength is required, an alloy of aluminum and 1.25 per cent manganese (3S) can be used. Butt, fillet, or edge welds can be made on these materials with entire freedom from heat or contraction cracks.

An alloy of aluminum with 0.25 per cent chromium and 2.5 per cent magnesium (52S), possessing even higher strength than the alloy of aluminum and manganese (3S), can be used at times. However, the material is more sensitive to the heat of the welding torch and difficulty is experienced in making other than butt or edge welds. Fillet welds or welds in gages heavier than $^1/_4$ in. may lead to production difficulties, which generally occur as cracks in the parent metal when the weld cools. These remarks apply only to torch welds, however, since such difficulties are not experienced with arc or resistance welds.

All of the alloys are produced commercially in various tempers in which the mechanical properties are raised considerably by applying predetermined amounts of cold work. The application of welding heat, however, anneals the material, and it follows that the strength of a welded joint is equal to the annealed strength of the base alloy. The annealed zone near the weld extends from three to five times the thickness of the metal on each side of the weld. Because of this fact, welded pressure-vessel design must be based on the annealed strength of the alloy. In other cases it is frequently possible to locate the welds so that advantage can be taken of the higher properties of cold-worked metal outside of the weld zone.

By suitably alloying aluminum with certain other metals, a class of materials has been developed in which the mechanical properties can be increased by heat-treatment. Two of these alloys are particularly suitable for welding operations. Aluminum alloyed with 1 per cent silicon and 0.6 per cent magnesium (51S) is widely used for welded furniture. Another similar alloy of aluminum consisting of 0.7 per cent silicon, 1.25 per cent magnesium and 0.25 per cent chromium (53S) is used for architectural and marine applications and aluminum beer barrels. Solutions of the constituents, and consequently the mechanical properties, are affected by the welding temperature. The heat-treated structure is not completely obliterated by the welding heat, however, and the strength of an assembly welded from these alloys is not predictable. The number, length, and type of the welds determine the extent of the annealing effect, and consequently individual tests of the assembly after welding are generally required to establish the strength accurately.

Gas welding is accomplished on all of these alloys with standard welding equipment, utilizing either the oxyacetylene or oxyhydrogen torch. Sound welds have been made with other welding gases, but when considering availability, welding speed, and general suitability, hydrogen and acetylene are most satisfactory.

Aluminum and its alloys are covered with an oxide coating

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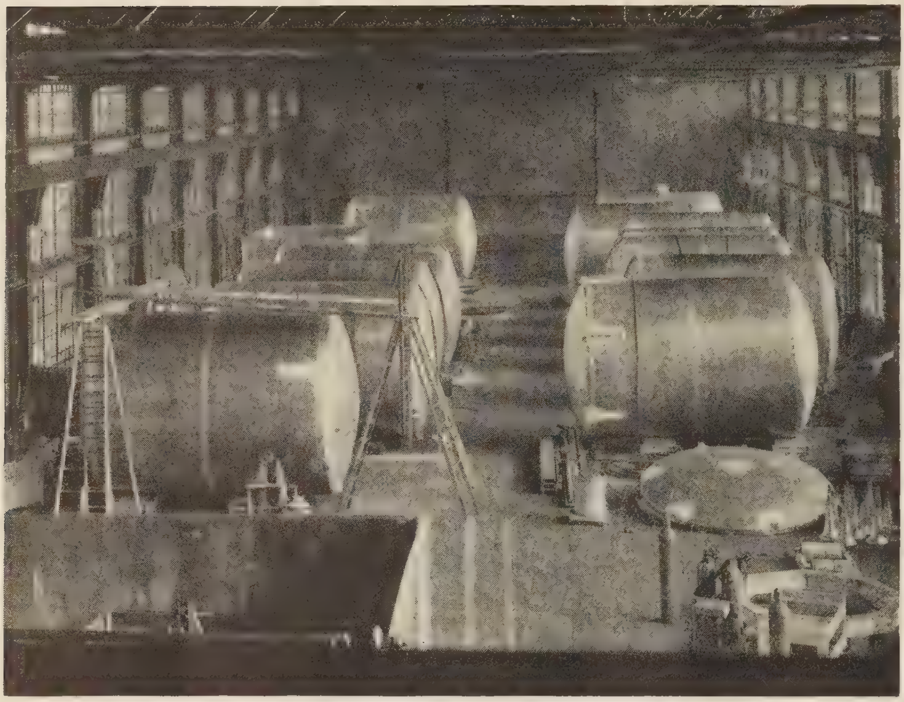


Shop Setup for Assembling 99.5 Per Cent Aluminum Beer Tanks by Torch Welding

formed in the atmosphere which prevents smooth coalescence of the weld metal unless a flux is used. A number of good fluxes are available which are generally mixed with water and applied to the welding rod by dipping. In this connection, it should be recognized that all of the welding fluxes are composed of chlorides and fluorides which, in the presence of moisture, attack aluminum. Residual flux deposits remaining on the joint after welding must be removed. Washing in boiling water or immersion for one half hour in a 10 per cent sulphuric-acid solution is effective in accomplishing this result. Complete removal of the flux from parts which are to be painted subsequent to the welding operation is essential since the action of the flux on the metal will lift the paint coating on the joint in a comparatively short period.

Proper preparation of joints is important in obtaining sound welds with good penetration. Gages $^3/_{16}$ in. and heavier should be V-notched prior to welding. Distortion and buckling can be held to a minimum by tacking the joint at intervals of from 8 to 12 in., or by the use of jigs to align the parts. In the latter case, jigs should be designed so that expansion and contraction of the parts can take place without putting a strain on the weld as the joint solidifies and cools.

The proper welding technique is a matter of practice and is not difficult to attain. Personnel familiar with the welding art on other metals are easily trained to weld aluminum when they are given an opportunity to examine the way in which the metal melts and the comparatively greater fluidity of the weld puddle. Qualification of welders is desirable, particularly on welds in heavy gages. Welds in material lighter than $^1/_4$ in. can be examined visually for quality with assurance that a smooth, regular bead will show a sound structure. On heavier material it is desirable to determine welding ability by having the welder produce a test plate on which the as-welded and reduced-section tensile strength, nick-break, and free bend tests can be made. Visual examination of welds in heavy gages is not a reliable guide as to the soundness of the weld structure. The details of such tests are outlined in section VIII of the A.S.M.E. code for unfired pressure vessels.

The brewery and dairy industries are finding extensive applications of welded construction. Storage and settling tanks, fermenters and yeast equipment for the brewery are all welded from aluminum with a high-purity grade of 99.5 per cent aluminum. Such tanks are welded with the torch using a filler wire of the same grade as the parent material. A smooth interior finish is provided on these vessels by hammering the weld bead flush with the metal surface. For packaging beer between the brewery and consumer, a beer barrel fabricated from the heat-treated alloy of aluminum, silicon, manganese, and chromium (53S) has been developed. This material has excellent resistance to corrosion and is equally as satisfactory as high-purity aluminum, in that it does not affect the taste or cause turbidity of the beer. In view of the fact that this alloy can be exposed directly to the beer, no pitching or coating need be applied to the inside of the barrel. It follows that for sanitary reasons no other method than welding can be used for assembling the barrels, and torch welds are used for all joints.

Materials for chemical equipment are subjected to rigorous service conditions and welded-aluminum construction finds many applications in this field. Most such equipment is made of the alloys 2S, 3S, or 52S, or the heat-treated alloy 53S because these materials are most resistant to corrosive conditions, and because the parts can be fabricated by welding. Specific examples of welded construction which have been successful, are to be found in equipment for handling synthetic nitric and acetic acids, hydrogen peroxide, cellulose acetate, and many of the chemicals used in the rayon industry. Welded aluminum is also used for handling distilled water. The fact that salts of aluminum are colorless has promoted the use of aluminum in processing and handling varnish and the raw materials used in its manufacture since no effect is apparent in the color of the product even though slight attack may occur. For the same reason, aluminum equipment is used in handling stearic acid as well as the nitric and acetic acids used in the rayon industry. Equipment for these purposes varies in size from small shipping containers to tank cars or large tanks 30 ft or more in diameter.

Widely varying service conditions in the chemical industry make it impossible to be more explicit in this field. It is suggested that specific problems in this connection be referred to an experienced manufacturer for recommendations concerning suitable alloys and design. It is also frequently desirable to expose typical joints to the exact service conditions to be encountered in order to provide reliable data for purposes of design.

The transportation of gasoline and oil in welded-aluminum truck tanks is well-established in practice. In this case the alloy 3S is used. Because of the light weight of aluminum construc-

tion, pay load increases between 30 and 50 per cent have been attained. Further improvement is anticipated with the use of the 52S alloy in which the higher unit strength should permit additional weight savings. A number of experimental tanks of this alloy have now shown good service records.

The alloy 53S is widely used for architectural applications where, because of its strength, excellent resistance to corrosion and weldability, it provides good appearance with economical fabrication cost. Doors and windows for buildings, buses, and railroad passenger equipment are made by welding this alloy. In the metal-furniture field, where light weight, durability, and sanitary construction are essential, the alloy 51S is used and joined by gas welding.

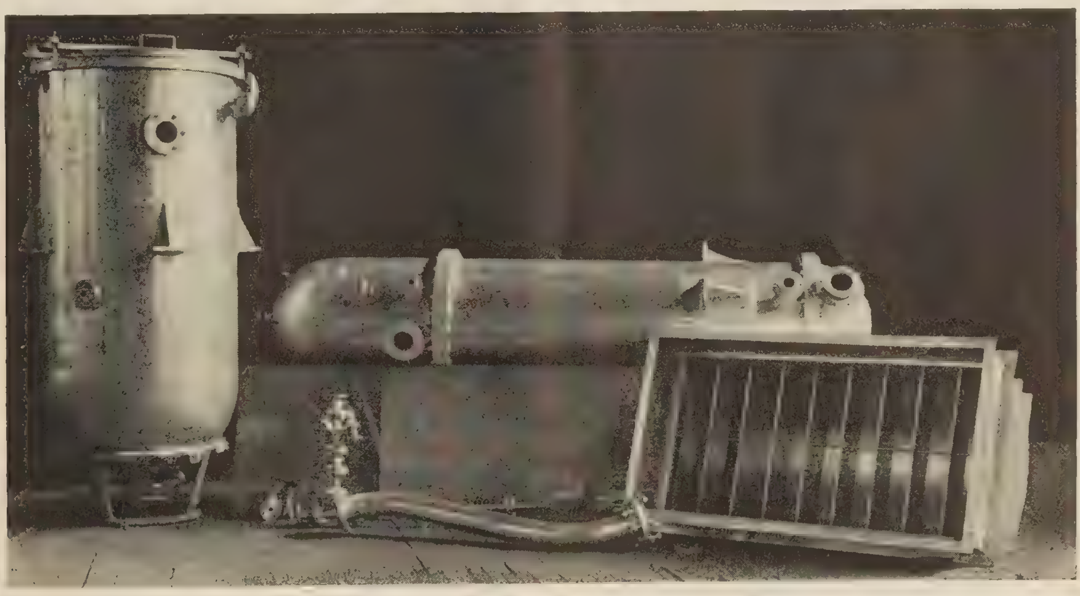
Assembling 53ST Aluminum Factory Sash With the Welding Torch

Metallic-Arc Welding

Metallic-arc welding of the aluminum alloys is gradually attaining more prominence in production operations, particularly for structural parts where expensive finishing operations are not required.

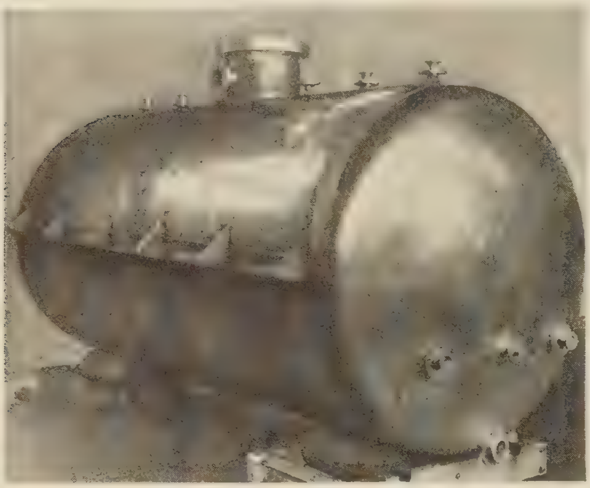
The process is particularly useful in minimizing distortion or buckling when welding aluminum. The high temperature of the arc fuses the parent metal rapidly with that deposited from the rod and a comparatively narrow zone in the parent metal is affected by the welding temperature. In spite of the high thermal conductivity of aluminum, the amount of expansion that takes place when arc welding is substantially less than when torch welding.

The rapidity with which the metal melts and freezes is also conducive to narrowing the annealed area near the weld in the cold-



Aluminum Equipment for Handling Turpentine Products Assembled by Torch Welding

worked alloys and to reduce the lowering of the heat-treated properties in the heat-treated alloys. While the area affected by the welding is smaller, it is not eliminated and the strength of the welds is substantially the same as the strength of gas welds. It follows that the previous remarks on designing welded structures on the basis of the annealed strength of the alloys applies as well to metallic-arc-welded parts.



Aluminum-Alloy (3S) Tank Made From Metal 1 In. Thick. Torch Welding Used for Fittings and Girth Seams. Metallic-Arc Welding Used for Attaching Supports

Special equipment is not required for arc welding aluminum. The standard d-c motor-generator sets can be used as well as a-c equipment, particularly the type which superimposes a high-frequency current in the arc circuit. Machine capacity depends on the gage to be welded. Table 1 shows suggested machine settings and rod sizes for average shop conditions.

Arc welding of aluminum can be done only with heavily coated rods. The flux coating serves to stabilize the arc as well as to remove the oxide coating thus permitting smooth coalescence of the weld metal. Rods must be dried thoroughly prior to use be-

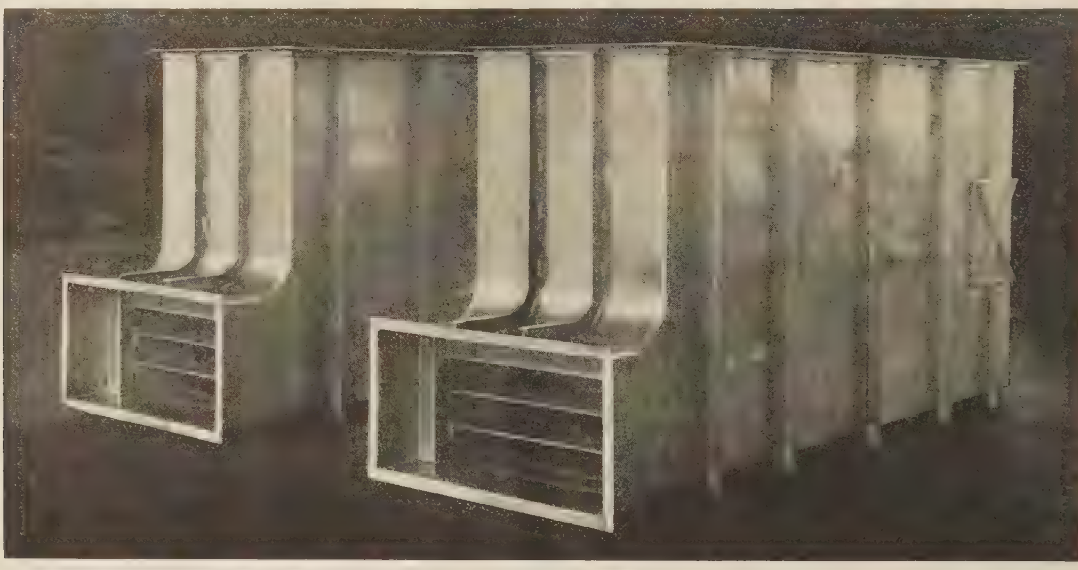
TABLE 1 ELECTRODE SIZE AND MACHINE SETTING FOR THE METALLIC-ARC WELDING OF ALUMINUM

Thickness, in.	Electrode diameter, in.	Amperes	Rods per lb
0.064	1/8	45–55	32
0.081	1/8	55–65	32
0.102	1/8	65–75	32
0.125	1/8	75–85	32
5/32	1/8 or 5/32	85–100	32–23
3/16	5/32	100–125	23
1/4	5/32 or 3/16	125–175	23–17
5/16	3/16	175–225	17
3/8	1/4	225–300	10.5

cause the formation of steam, if water is present, causes excessive spatter. Flux coatings on the rods are somewhat hygroscopic, and if the rods are stored for periods longer than a week before using them, it is advisable to dry out the coatings by baking the rod for several hours between 200 and 300 F.

The fluidity of the weld metal when making butt welds on aluminum requires the use of a backing strip to confine the penetration bead. Backing strips are generally made from copper or steel and grooved about 1/16 in. deep and 1/2 in. wide to provide enough molten metal on the bottom of the weld to prevent "piping" from the solidification shrinkage in the weld bead. It is difficult under shop conditions to make butt welds on gages lighter than 0.081 in., and fillet or lap welds on gages lighter than 5/32 in. These limitations have been established because of the difficulty of controlling the arc on the lighter gages. In addition, the rapid freezing of the weld entraps gas in the cast weld zone, causing a porous structure. No upper limit on thickness has been established and sound welds in plate 2 in. thick have been made.

The preparation of joints for metallic-arc welding is simpler than the preparation of the joints for torch welding. When making butt welds, material up to 1/4 in. thick requires no preliminary V notching or other preparation. Heavier material should be Veed to within 1/4 in. of the bottom of the section. Good penetration can be obtained on material from 1/4 to 1/2 in. thick without edge preparation in those cases where a weld bead can be laid down from both sides of the section.



Air Ducts With Side Walls Stiffened by Metallic-Arc Welding the Channel Sections to Sides

The welding flux used for coating rods is similar to the torch-welding flux in that it should be removed from the joint after welding. The same methods already mentioned are applicable in

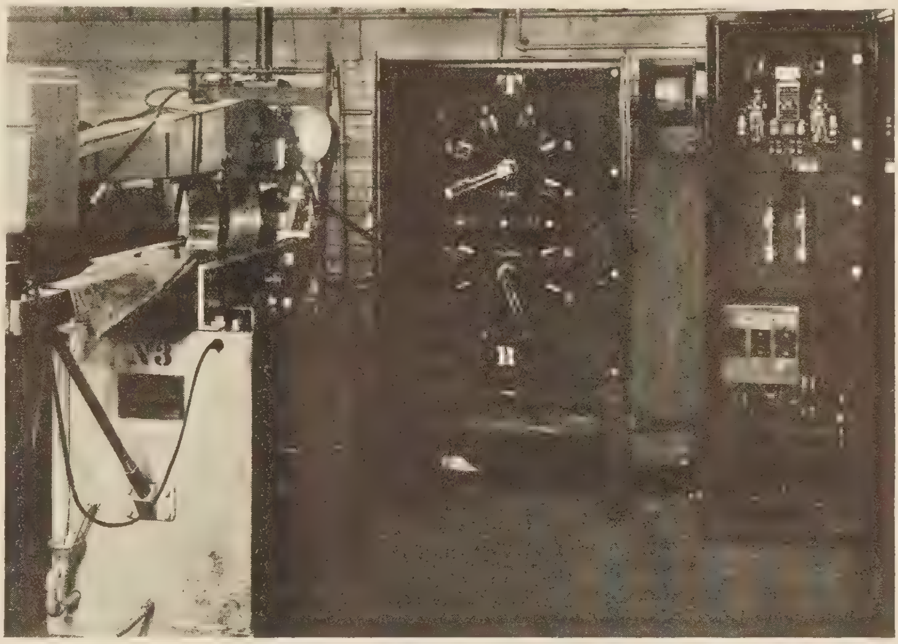
this case though, and because of the rougher surface of the weld, vigorous application of a wire brush is desirable to insure complete removal of the flux.

Electric-Resistance Welding

Substantial progress has been made in the past two years in the development of suitable equipment for spot and seam welding aluminum. This result has been brought about largely by the demands of the aircraft industry for a more economical method of assembling all-metal airplane structures than the driving of thousands of small rivets. Its use, however, has not been limited to aircraft and is extending to other fields concerned in the assembly of sheet structures.

While the process is just reaching the production stage, the development of equipment and investigational work on strength and resistance to corrosion of spot welds have been conducted for several years.

With few exceptions, equipment as designed and used for spot welding other metals is unsatisfactory for use with the aluminum alloys. A tabulation of the characteristics required for suitable aluminum-alloy work, will tend to emphasize the reason for this condition.



Equipment Setup for Production Spot Welding of Aluminum Alloys

Spot-Welded Rack Assembled From Aluminum-Alloy (53ST) Extruded Shapes

Spot and seam welding of the aluminum alloys is essentially a precision job, but, if the equipment provides the following essential features, consistent results can be obtained with any of the aluminum alloys. These features are:

1 Precision synchronous timing for the highest quality strong alloy work, or reasonably accurate nonsynchronous timing for certain jobs using common alloys.

2 A proper value of welding current for the work in question.

3 Suitable electrode design.

4 A quickly adjustable and accurate electrode pressure mechanism.

5 A design of machine to minimize electrode hammer blow.

6 Equipment to insure that the adjustments as made are maintained for successive duplicate welds.

In spot welding aluminum, a spheroidal cast bubble-shaped area is formed at the interface between the parts by the passage of the current. Experience has determined that the depth of the bubble should extend about two thirds of the distance to the sheet surface to obtain consistent strength and best resistance to corrosion in the welds. Accurate control of the current and the time is essential to obtain consistent results, particularly on the heat-treated alloys. Electronic timing equipment is now available with adequate capacity to handle the relatively heavy currents for short times that are required for welding aluminum. The time and current required vary with the gage, and approximate values are shown in Table 2.

TABLE 2 APPROXIMATE MACHINE SETTINGS FOR SPOT WELDING ALUMINUM

Thickness, in.	Cycles	Approx amperes	Electrode pressure, lb
0.020	1	16,000	300– 500
0.036	2	20,000	400– 650
0.064	4	24,000	500– 800
0.081	6	28,000	600– 900
0.125	9	35,000	700– 1200

Note: Cycles based on 60-cycle power supply.

It should be pointed out in connection with item 2 that the conventional kva rating on a machine nameplate is generally not a reliable means of establishing the capacity of resistance-welding equipment. The current at the welding tips is varied by the distance between the arms and by the throat depth so that two machines of exactly the same kva rating may have substantially different capacities. Insufficient heat to make a weld cannot be compensated for by increasing the time when welding aluminum. The capacity in amperes for the various arm positions on a machine can be determined easily by simple electrical tests, and a calibration of the equipment is the first essential when starting on the welding of aluminum.

Proper electrode design and maintenance are essential to obtain consistent spot welds on aluminum. Variation of the con-

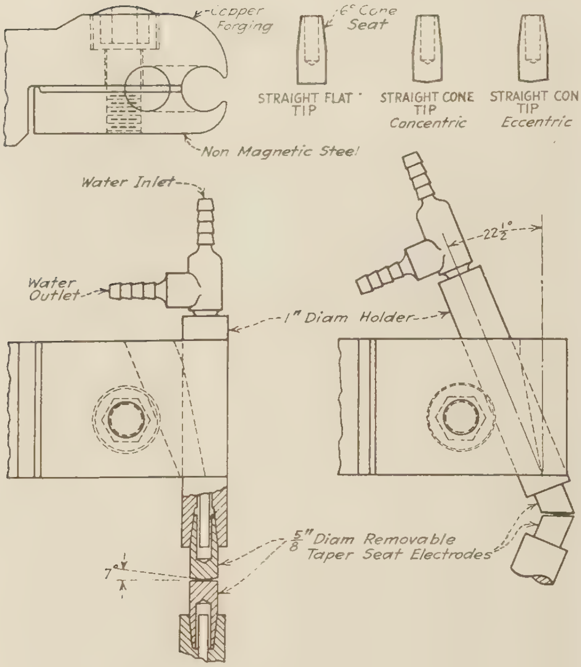


Fig. 1 Spot Welding Tips and Holder

tact area of the electrode tip will change the current and pressure distribution and contribute to inconsistent strength of the welds. The common method of dressing electrodes with a file is entirely unsuitable for spot welding aluminum. A simple means maintaining tip contour is shown in Fig. 1. A 165-deg included-angle cone is machined on the tip and when it is in use no cleaning is done on the tip surface with anything but a very fine grade of abrasive cloth. A tip of this type will make between 250 and 400 spot welds before remachining is necessary.

The other features listed concerning electrode pressure mechanism and hammer-blow, as well as equipment to insure the duplication of electrical conditions for successive welds, are questions of machine design. The welding-machine manufacturers are familiar with these requirements and can supply equipment to meet any desired application of spot or seam welding of aluminum.

Strength of Spot Welds

The strength of spot welds in the aluminum alloys has not yet been investigated thoroughly. The shear strength of spot welds has been determined for a limited number of gages and alloys, and typical results are shown in Table 3. Because of the rapid changes in the equipment and process, it has not been possible

TABLE 3 BREAKING LOADS ON SPOT WELDS IN VARIOUS ALUMINUM ALLOYS

Alloy	Breaking load, lb per sq in.
2S½H	200
52SO	240
52S½H	450
Alclad 17ST	350
Alclad 24ST	360
53ST	350

Note: These figures represent the average shear strength per spot obtained on five or more 1-in. wide two-spot test specimens. All specimens were made from sheet 0.032 in. thick.

to cover the entire field of gages and alloys. However, further testing is being done and where specific data not yet determined are required, they can be obtained on relatively short notice.

Some work has also been done in ascertaining the fatigue strength of spot welds. No absolute values of fatigue strength have been determined since the tests were designed to compare spot-welded with riveted joints. Results of this work to date indicate that properly designed spot-welded joints withstand alternating or vibratory loads as well as riveted joints. When changing from a riveted to a spot-welded joint, tests of different joint designs are advisable to determine the most efficient. Aircraft gasoline tanks which must withstand a 25-hr vibration test are built with seam- and spot-welded joints.

Resistance of Spot Welds to Corrosion

Tests on spot welds exposed to salt spray, the atmosphere, and to alternate immersion in tidewater from the Hudson river, have been conducted for the past several years.

Spot-welded specimens have been exposed for one year on the coast at Point Judith, R. I. The exposure conditions in this locality are very severe because salt spray is deposited on the specimens in stormy weather. Tension tests on the exposed specimens show no significant losses over this period on 2S, 3S, 52S, 53S alloys, and on all of the Alclad forms of the aluminum alloys. These results confirm earlier laboratory salt-spray tests.

Spot-welded boxes simulating aircraft construction, have also been exposed to alternate immersion in Hudson river tidewater at Edgewater, N. J. The results indicated that the high purity coating on Alclad 17ST and Alclad 24ST alloys, protects the spot welds just as effectively as rivets or sheared edges are protected. On the basis of the tests run to date, the resistance of spot-welded joints to corrosion is comparable to that of riveted joints in the same alloys.

Casting or Welding in Machine Design

By J. L. Brown,[1] East Pittsburgh, Pa.

This paper is a presentation of the problems involved in choosing between a cast machine part or a part fabricated by welding for any specific purpose. The author discusses the advantages and disadvantages of each method in the light of cost, utility and appearance of the finished product and compares the casting method with fabrication by welding for several specific structures. Fields in which fabrication by welding can and cannot compete with castings are also mentioned, together with problems involved in designing a welded part to replace a casting.

Castings have been made for hundreds and even thousands of years. The art of producing structures by welding is a distinctly modern development, and in spite of the fact that extensive application of it has taken place only within recent years, it has expanded to fill a large and rapidly broadening field of usefulness. The question often arises as to what extent welded structures can economically replace castings.

The use of welded structures is largely dependent upon another development of modern times, the rolling of steel. A section of rolled steel does not in itself ordinarily form a complete structure but depends upon some means of attaching it to other portions of the structure. For this purpose it has been the custom for many years to use small wrought-steel parts such as rivets or bolts thus departing entirely, in such structures, from the casting method of fabrication.

Broadly speaking, the casting method of fabrication is essentially the flowing of molten material into any given location and permitting it to solidify. Ordinarily some particular shape of the solidified material is required and this end is accomplished by flowing the molten material into a mold previously prepared to give the desired shape. In certain cases it has been found desirable and feasible to attach to the casting a part such as a lifting eye or screen for a ventilating opening which remains in the solid form during the casting operation by suitably placing it in the mold, with the portion to be joined to the casting extending into the space provided for the molten metal. Another well-known practice similar in principle is the joining of trolley rails by thermit welding. In this construction, the familiar bolt and fishplate joint is replaced when the operation is complete by a mold-formed block of cast metal firmly attaching to the ends of adjacent lengths of rail.

In some of these cases the solidified metal holds the associated part by closely following its contour and grasping it firmly without actually bonding to it. In other cases the heat of the molten metal has been sufficient to melt the surface of the solid parts so that when cooling, the casting and the part cast in it become bonded together in one solid mass, the part set into the casting retaining its own peculiar shape external to the casting and the casting taking the form of the mold. This type of work is illustrated in Fig. 1.

Compare with this the deposit of weld metal joining the ends of a pair of rolled-steel bars scarfed out to produce a flush weld, also shown in Fig. 1. The V-shape of the scarfed joint has formed what is substantially a mold to receive the molten weld metal. The latter cools and forms a V-shaped casting. The heat applied has melted not only the weld metal but the adjacent surface metal of the steel bars. The result upon cooling is a solid

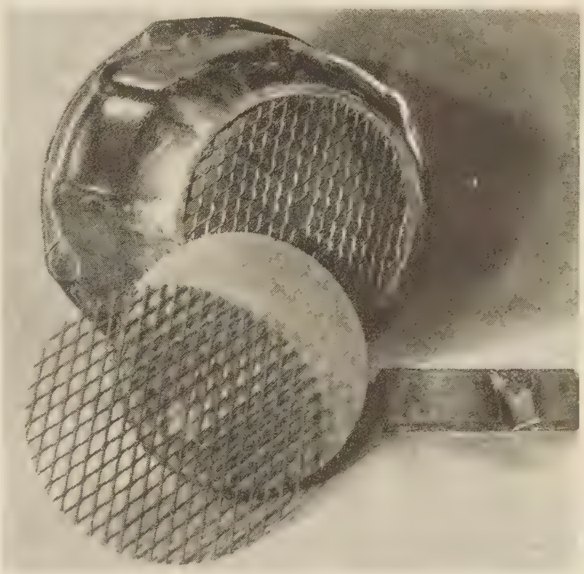


Fig. 1 Expanded-Metal Screen Bonded in a Casting

(The screen, the screen in the core sand, and the screen bonded in the casting are shown at the left. Two steel bars butt-welded together are shown at the right. The cast-in screen and the butt-welded bars have the same essential elements, that is, parts of unmelted rolled steel joined together and to the bonding material by molten metal which has subsequently solidified.)

structure consisting of two rolled-steel bars joined by a casting. To be sure, the volume of metal cast into the joint is a small percentage of the weight of the entire structure, and the conventional mold usually associated with castings is absent because it is not required and the melting and solidifying process is progressive across the joint instead of being a bulk operation. These differences as compared with conventional casting practice are, however, adaptations to suit the conditions, and permit the maximum possible convenience and efficiency in producing the complete structure.

Not only is the deposit of weld metal at the joint of a welded structure essentially a casting, but all sections of rolled steel were originally in cast form, that is, cast as ingots from the steel furnace. The rolling process lengthens and widens such castings, and at the same time reduces the thickness. A welded struc-

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ture therefore becomes a casting stretched out by rolling; cut, and assembled to give the required form; and locked in position by relatively small portions of cast metal appropriately placed and attached by the welding process. The question of weldings versus castings therefore resolves itself into how far this stretching and piecing together process should be carried in displacing the method of making the original casting to the final form in one integral piece.

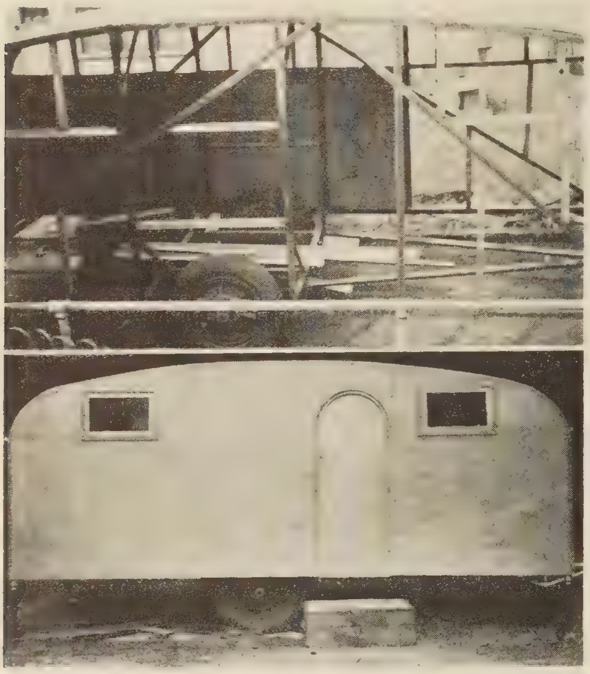

Fig. 2 Welded Automobile Trailer
(Top: The welded-angle rib structure of the trailer. Bottom: The completed trailer.)

Many of the advantages to be gained by the welding method of fabrication are closely associated with and dependent upon the advantages of rolled steel. Since this material is produced in tremendous bulk by highly developed high-speed machinery and, in a limited number of standard shapes and sizes, it is available at relatively low cost per unit weight or strength. Relatively thin sections of great length and of such shape as to give most efficient use of the material are readily produced without reference to the thickness of section required for flowing of molten metal or to resist shrinkage stresses such as arise in the solidifying process in the case of castings. Furthermore, the manufacture of structures of a variety of shapes and sizes and for any and every purpose may be provided for by carrying a stock of a limited number of shapes and sizes of rolled steel and supplying some satisfactory means of cutting them to size and joining the pieces in assembly.

Fabrication by Welding

At this point the advantage of welding as a manufacturing process becomes apparent. When the parts of the structure have been sawed, sheared, punched, flame-cut or otherwise made to shape and subsequently arranged in proper relative position, the joints may be formed by welding with a speed and efficiency impossible to equal by any other known method of joining steel to steel. When, however, this has been done the question still arises: "Should this structure have been a casting?"

Experience has shown that in many cases this question is answered correctly in favor of a welded design, and this answer will be more or less obvious depending upon conditions. For example, it is obvious that very large structures which are assembled from many pieces at their final destination must be of rolled members joined together by welding or some other method. In other words, no one would suggest substituting cast members for rolled-steel members for the frame of a large bridge or office building. Here the advantage of low-cost standardized shapes with efficient distribution of material is realized and the efficiency of fusion welding as a means of producing the joints of such structures has been demonstrated repeatedly in projects completed in recent years.

Even though the structure be of much smaller size than the type just suggested, but having the same general character wherein long members of relatively thin section predominate, thus giving a low ratio of cubic volume of material to cubic volume of the structure, castings will be found unsuitable for the reason, among others, that molten metal will not flow through long thin passages in the mold without becoming cooled beyond the point where fluidity disappears.

The author recently used about eight hundred pounds of $1^3/_4 \times 1^3/_4 \times {}^3/_{16}$-in. steel angle welded into a framework for the 16-ft house trailer shown in Fig. 2. It would certainly be hopeless to consider making this structure in the form of one or more castings not only because of the considerable increase in section of the material required for flowing of the molten metal and correspondingly great increase in weight, but because of the lack of the elaborate pattern-shop and foundry facilities required, and the expense involved in producing a structure of this size and shape by the casting method.

Where box-like structures of this character are to be produced in a single casting, large elaborate and expensive cores are necessary, and hazards and delays are introduced. A job of this type falls easily into the welded class, where available standard shapes can be quickly cut and fabricated by welding.

The speed with which welded structures may be produced from standard shapes carried in stock often makes this the preferred method to fill a given order even though a casting might be equally satisfactory for the purpose and indeed the casting method may be so advantageous for the part in question as to be adopted for producing it for subsequent orders. Demand for early delivery, therefore, often makes it obvious that the part must be of welded steel. If the part is not to be reproduced, the casting will still be less attractive as the cost of the pattern has to be added to the cost of the casting and charged to the order. Even the prospect of infrequent demands for the part in the future might make a pattern undesirable because of the problem of pattern storage and repairs to it due to wear and tear resulting from handling and weather conditions.

Among other general considerations influencing the choice of welded or cast designs, are the complexity of the structure, the amount of machining, and the appearance requirements. One of the advantages stated for the use of rolled steel was the availability of standard shapes at low cost. If, however, the character of the structure to be produced does not lend itself to the use of these shapes without a large amount of forming and piecing together, as is the case with arc-welded frames shown in Fig. 3, it may require a much closer analysis to decide whether a cast or welded design will be the more advantageous. Welded designs have their greatest advantage when the pieces used in making them are few and simple and are held in assembly with a minimum of welding. When structures, especially small structures, become complex, the cost of a welded design might substantially exceed the cost of a cast design.

If in addition to the fact that a suitable structure can be pro-

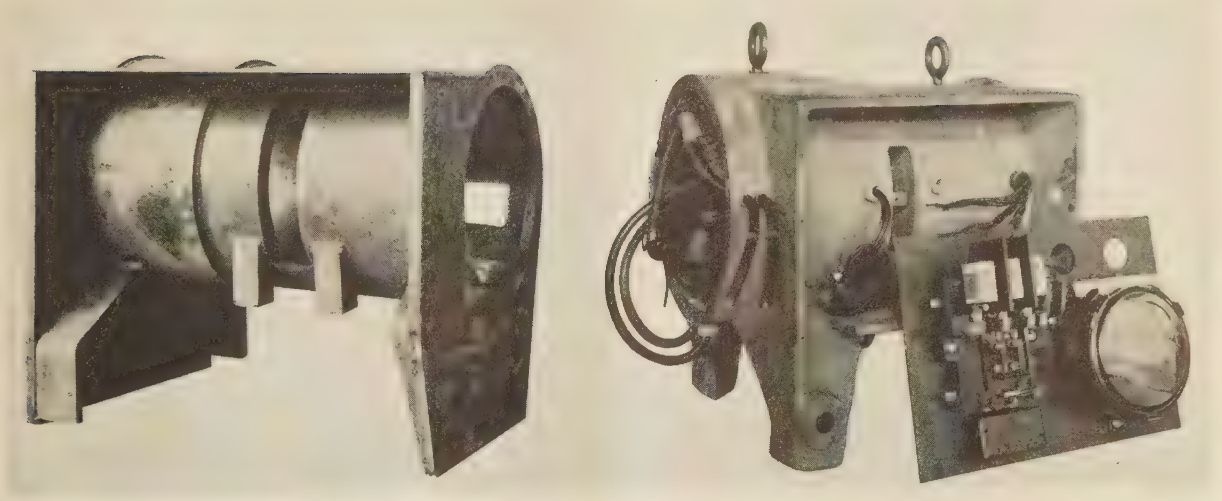

FIG. 3 WELDED AND CAST ARC-WELDER FRAMES
(The welded-steel frame at the left has so far been unable to compete in cost with the cast frame shown at the right.)

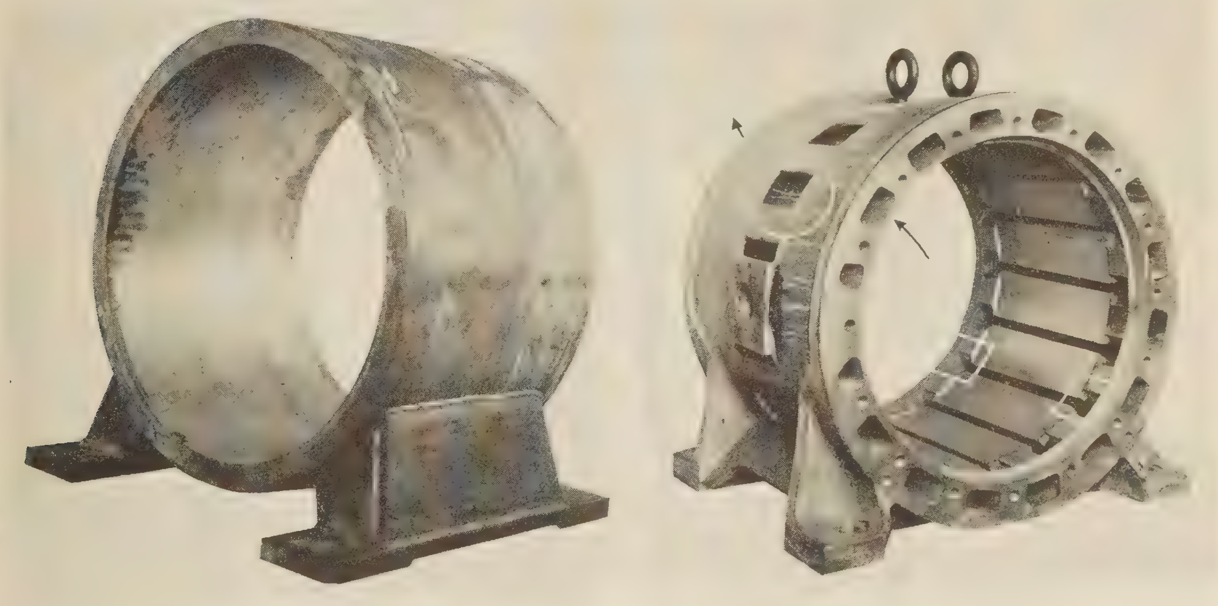

FIG. 4 WELDED AND CAST MOTOR FRAMES
(The motor frame at the left is correctly designed as welded. The pieces used are easily formed to shape, and the area of the frame section is required for magnetic purposes. The cast motor frame is of intricate shape but can be cast easily. In the case of the casting there are no magnetic requirements and the laminated stator is subsequently pressed in and serves all magnetic purposes of the frame assembly.)

duced by judicious choice of a few simple rolled-steel parts joined by a few advantageously placed welds, the parts can be so placed as to eliminate the necessity for any considerable amount of machining, the casting method will usually give place to the welding method of manufacture.

Finally, if the character of the structure is such that the distinctly angular appearance resulting from the use of standard rolled sections is not acceptable and the number required is so small that the expense of tools required for forming the constituent parts to a more pleasing shape is not justified, the tendency will be in favor of the casting method. However, where appearance is of less importance and the requirements of the structure may be fulfilled by welding together a few simple pieces of angle, channel, H-beam or flat material, as illustrated in Fig. 4, the designer will usually be strongly influenced to take advantage of the welding method.

When the application of the foregoing general analysis to the problem does not give an obvious answer, a more detailed and careful analysis of the facts which enter into it must be made. For example, we may ask: "How should a motor frame be made?" Depending upon the various conditions the answer is given, and may be correctly given, sometimes in favor of the casting and at others in favor of a welded design. What these conditions are and how they influence the final answer as to

whether a cast or welded design will be the more advantageous will be discussed in the following paragraphs.

In the case of castings, the availability of a variety of materials having widely differing characteristics permits a choice of that one having the greatest net suitability for the particular purpose. For welded structures, on the other hand, the choice of materials is more narrowly limited, so that in many cases a greater compromise must be made. Whereas for general purposes welding and especially arc welding is largely confined to steel, castings are made in large volume from not only steel but from the large variety of cast irons, semisteels, and nonferrous materials. Each of these is particularly suited to a purpose or purposes and because of their adaptability in fulfilling these purposes it is particularly difficult to displace them by a welded-steel substitute. However, in seeking to make a substitution, it is necessary to consider the principal advantages of these cast materials.

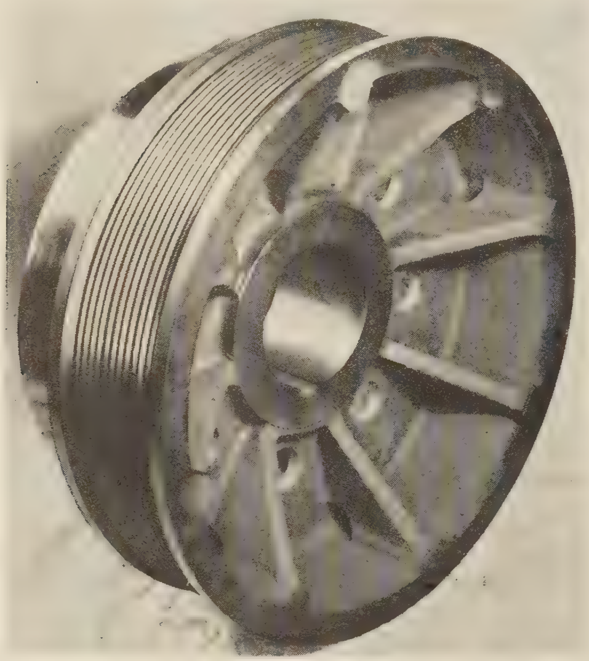

Fig. 5 Rope Sheave Machined From a Casting

Advantages of Cast Materials

Cast irons and semisteels comprise by far the greatest bulk of all castings made. Their texture provides ideal wearing surfaces for such items as brake drums, rope sheaves similar to the one shown in Fig. 5, and cylinders for internal-combustion engines. They are resistant to corrosion and may be produced with fair magnetic properties. Cast irons are used for a great variety of work where stresses are low, either because forces are small or the section of material is ample, or where ease of machining is a factor. Semisteels provide up to double the strength of ordinary grey cast iron while maintaining the characteristic cast-iron structure so advantageous for machining and other purposes. All metals of this class, however, are very low in ductility and they do not lend themselves readily to joining to other parts by welding.

Malleable-iron castings are produced in comparatively small volume. They are used for relatively small machine parts where freedom from brittleness is important and where steel castings would be more difficult to make on account of the thin sections employed and would be stronger and more expensive than necessary. They machine easily, have fair magnetic properties and like cast-iron castings are not suitable for joining to other parts by welding. They are more expensive than cast iron and take much

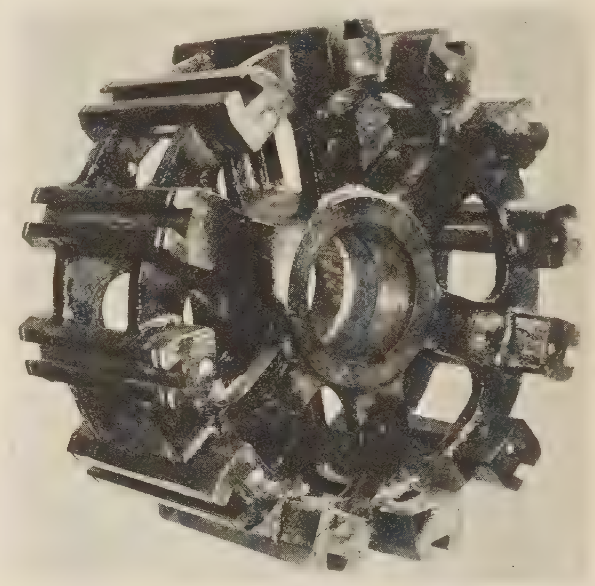

Fig. 6 Cast-Steel Spider

(This cast-steel spider, carrying a large weight of segmental laminations at high angular velocity, is provided with adequate cross section at the junction of axial dovetail supports and radial arms. In a welded spider it is difficult at such joints to obtain adequate cross section of weld to resist the high centrifugal forces to which the spider is subjected.)

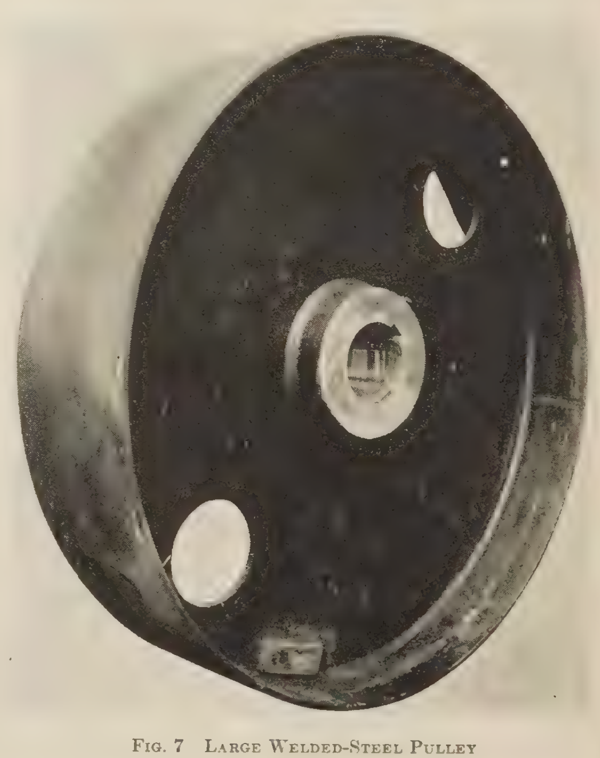

Fig. 7 Large Welded-Steel Pulley

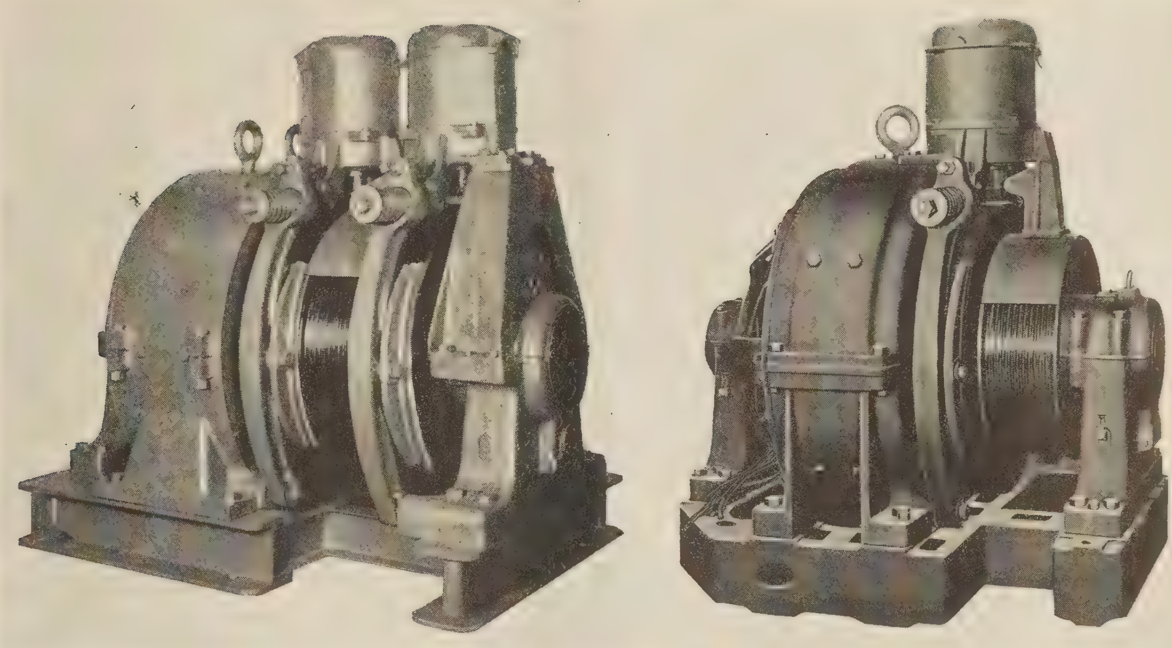

Fig. 8 Cast-Iron and Welded-Steel Bedplates for Elevator Motors
(Dissimilar in appearance but serving the same purpose.)

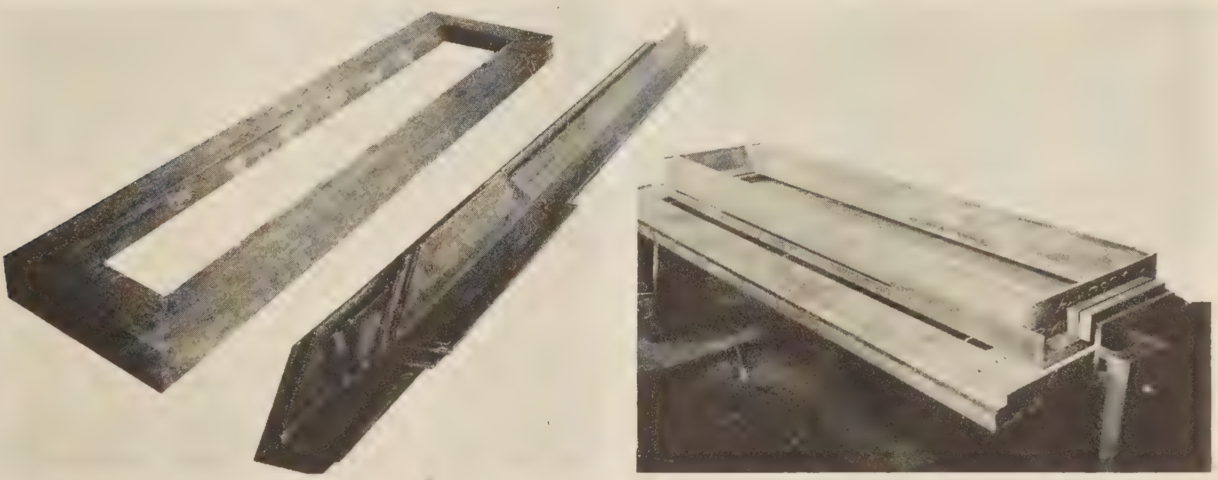

Fig. 9 Welded Bedplate Made From Sheared Angles Welded Complete in One Position
(Left: The completed bedplate. Center: The sheared angles. Right: Angles mounted in jig with welding completed.)

longer to produce on account of the long oven treatment which is an essential part of their manufacture.

Steel castings such as shown in Fig. 6 are used chiefly for highly stressed parts or parts subjected to shock loading, and where a high-permeability magnetic material is required or where it is desired to attach them by welding to other parts of the structure. They are more difficult and expensive to make than the other ferrous-metal castings for various reasons. They are less resistant to corrosion but are of high ductility.

Nonferrous-metal castings are used where light weight is required, where corrosion resistance is important, where the advantages of die-casting or plastic forming are sought, for bearing surfaces, and for electric or heat conductivity. Welded structures have been at such a disadvantage in these fields that they have not ordinarily been considered as important competition. However, more recent developments in the welding of stainless steel have resulted in intrusion of welded structures into the light-weight and corrosion-resistant fields previously considered as belonging exclusively to nonferrous metals.

Welding Versus Casting

It is, of course, in the fields heretofore considered as belonging to ferrous castings that the welded-steel substitute has gained most ground and in very many cases important savings may be

made by substitution of a carefully considered welded design for what would otherwise be a ferrous casting. The welded-steel product may be designed to have the strength of the steel casting. It will slightly exceed in its rolled-steel members the ductility and permeability of the steel casting and where rolled-steel sections may be adapted to the required shape of the structure it may easily be designed for a considerable saving in cost.

Where corrosion resistance is required, where ideal wearing surfaces are necessary and where large quantities of irregularly shaped parts of the type shown in Fig. 4 must be produced and subsequently machined, it will be difficult for the welded steel to compete with the cast-iron or semisteel product. However, where cast iron has been used for other purposes such as bed-plates, sole plates, large machine frames, large pulleys, fly-wheels, jigs and fixtures, a healthy competition exists between the casting and welding methods of fabrication. This is illustrated in Figs. 7, 8, and 9. When it is possible to reduce the structure to an assembly of simple shapes or where equipment is available for preforming the parts the advantage will usually lie with the welding method. However, in proportion to the irregularity of the surfaces required expressed in terms of standard rolled shapes, often in proportion to the activity of the part and inversely proportional to its size, the advantage will lie with the casting.

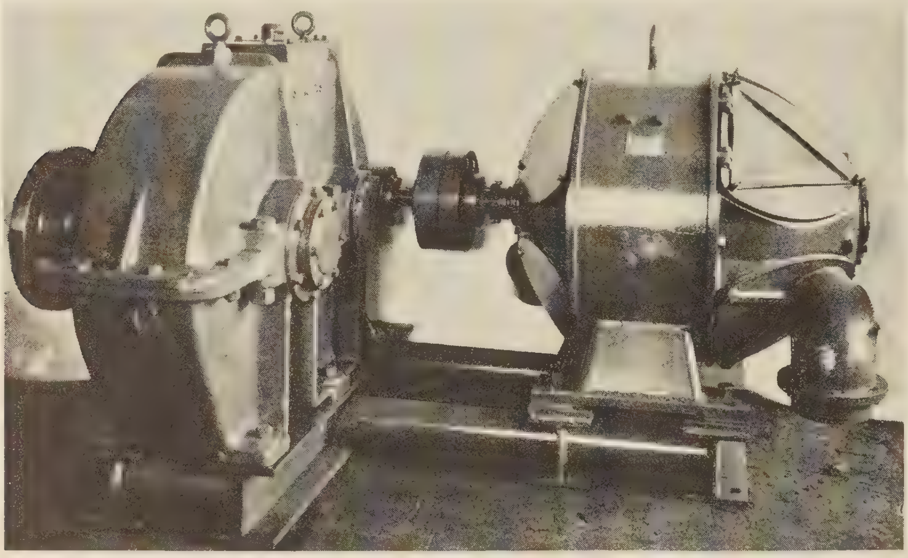

Fig. 10 Dissimilar Shapes Used in a Bedplate Which Because of the Shapes Used Have a Displeasing Appearance

Analysis of a Welded Design

In analyzing a particular case where the solution is not obvious, it is recommended that a casting be designed first and that it be designed to secure the greatest efficiency in appearance, utility, and cost. Obviously, it will be useless to design a welded structure to compete with a poor cast design. The result may be a much less efficient welded structure than a competitor's cast design, which may prove serious. Also, it may be pertinent to state that it is not safe to assume all existing cast designs are efficient designs and therefore safe criteria to use in making competing welded designs. There have been many welded designs made which were cheaper to produce than a replaced casting but which in turn could be replaced by a still cheaper cast design.

Having made the best possible cast design, and ascertained the probable cost as produced with foundry equipment consistent with the estimated activity of the part, we may proceed with the design of the proposed welded substitute.

The first thing to remember in this connection is that the most efficient welded structure will probably be quite unlike its cast counterpart in appearance. Unlike the casting, it is not hampered by requirements of section for flow of molten metal and to resist or avoid shrinkage stresses, or of shapes to facilitate withdrawal of a pattern from the mold. On the other hand, it will be hampered by the limitations of standard rolled shapes and the necessity of placing welds in such a position and manner as to avoid distortion and rupture upon cooling. The pieces of steel comprising the assembly should be so chosen and placed that the structure will serve some definite function rather than adhere to some preconceived appearance. By no means should we understand that all welded structures are of poor appearance. Skill in design should result in a good appearance but it will be at the same time an appearance consistent with the character of the rolled shapes available, rather than an exact copy of the cast shape most desirable for the purpose.

The problem of cutting the pieces is an important one. The means available are usually one or more of the following: Shearing, punching, flame-cutting, and sawing. Shearing and punching are ordinarily the least expensive. Flame-cutting is the most flexible method of shaping steel parts for welding and requires a minimum of equipment. However, it will usually be considerably more expensive than shearing or punching, not only because of the time consumed in the cutting operation, but also because the burned edges require subsequent cleaning from cinders preparatory to welding. The cost advantage of shearing will result in a preference for flats and angles which lend themselves more readily to this method of shaping than the more complex shapes such as channels, I and H beams.

The design should be so made that as far as possible all welding will be confined to ends and edges of the parts and in such a way as to secure symmetrical placement of the welds so that shrinkage of one will be balanced by shrinkage of the other, thus tending to avoid distortion. The problem of weld shrinkage with danger of rupture as well as distortion is a very serious one and it is sometimes necessary to peen the weld to relieve the stresses set up or to anneal the whole structure. In fact, for rotating parts, especially those rotating at high speeds, annealing after welding is the rule rather than the exception.

When the structure becomes complex, and often in the case of simple structures, it becomes necessary to provide some sort of fixture to hold the parts in assembly while welding is taking place. Furthermore, the parts must be placed and held in the fixture each time one of the structures is made. It is obvious, if all other things are equal, the fewer the number of pieces the smaller the cost, because of a smaller amount of welding and less time consumed in cutting and arranging parts preparatory to welding.

This phase of the problem is sometimes modified by the fact that when pieces become fewer they usually become larger. If the pieces of the larger number are of such size that a man can handle them without the aid of a crane and the pieces of the smaller number are of such size that a crane must be used in placing them, it may prove to be less expensive to use the large number of small pieces.

The cost will be less, all other things being equal, if the design is so arranged that all welds may be made without the necessity of turning the partially welded structure to another position. If two or three crane lifts are necessary in the course of the welding operation, the cost will be increased appreciably. This, therefore, should be avoided as far as possible.

In very many cases, it is unnecessary to weld a long seam or joint continuously. The character of the rolled-steel parts will usually be such that a series of short welds will give 90 per cent of the strength of the structure which would be obtained by a continuous weld. Also, the shrinkage and distortion encountered with the continuous weld will be avoided. In some cases, it will be found helpful to weld a little and permit the weld to cool, then weld a little more and cool, thus preventing a great accumulation of heat and reducing the shrinkage of the parts correspondingly.

In welded design there are a number of considerations such as those discussed in the preceding paragraphs which, though easily overlooked, have a vital bearing on the competitive cost of fabrication by welding with respect to castings. If they are neglected the cost will doubtless be higher than the competitive cast design, and if further the criterion of a skillful cast design is not set up, the result will be a structure made by distinctly modern methods but probably distinctly lacking in economy.

The same careful attention to details urged in seeking economy in producing welded structures will result in avoiding a monstrosity in appearance. The easiest thing to produce by welding is what is known in the vernacular as a "christmas tree," that is, all kinds of pieces of dissimilar shape attached together in one assembly, but having, as far as the eye is concerned, no logical relation to each other. The end view of an H or I beam looks like an H or an I, respectively. The side view of such beams has a concave appearance while the top and bottom are flat surfaces. The side of a pipe or round bar has a convex appearance. When these dissimilar shapes are used in a welded structure as, for example, the bedplate, Fig. 10, they present a displeasing appearance because of their lack of uniformity and symmetry.

Avoid as far as possible placing welds in conspicuous places. They are rough and irregular at best. Welds are sometimes filled with iron filler and sanded, but this represents additional expense and filler will crack off if bruised. Welds made with coated rods are in general smoother than those made with bare rods and bare-rod welds laid with the automatic welder are smoother than hand-laid welds. Avoid exposing rough, burned, or sheared edges. Corners formed by the heel of an angle, for example, are more pleasing than burned or sheared abutted edges placed in a similar relation.

It may be that the best appearance and the greatest efficiency in cost and utility can be obtained by combining the use of rolled steel and cast steel in the same structure using welding to join not only the pieces of rolled steel together but to join them to the casting also. This is very simple when cast steel is used, but, because of physical changes which occur in cast iron when melted by the welding, it is not nearly as practical to join cast iron to steel. It is possible to join these materials by the use of certain kinds of bronze welding rods or welding rods of other materials which melt and adhere to the surfaces of the parts to be joined without the necessity of development of sufficient heat to melt the cast iron. The structure of the cast iron is thus not interferred with and the joint may be made as strong as the cast iron itself. Certain difficulties sometimes arise in connection with the use of this method, however, on account of accumulation of heat resulting in warpage.

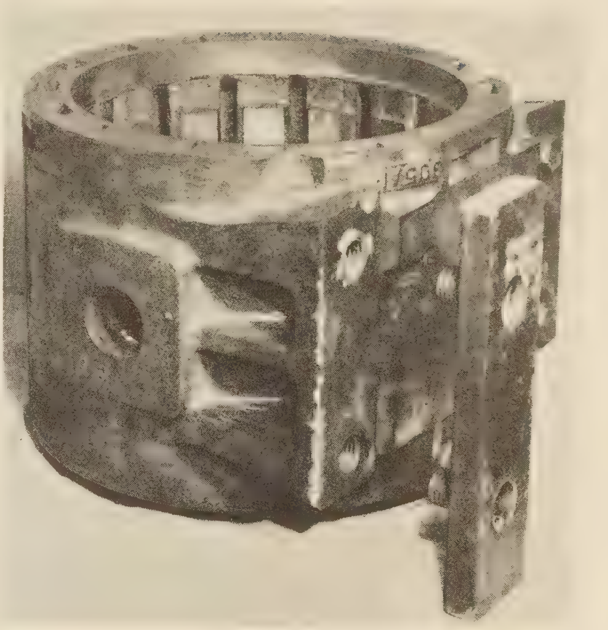

Fig. 11 Cast-Iron Frame With Steel Feet Welded to Studs Threaded Into the Frame

An interesting method of attaching steel to cast iron has been used successfully in which the rolled steel member is not welded to the cast iron directly but is welded to the projecting ends of large steel studs threaded into the body of the cast iron, as shown in Fig. 11. No melting of the cast iron takes place. The welded joint from stud to rolled-steel member is quickly made by the electric arc. The body of the cast iron does not become heated appreciably so that warpage is avoided. The welding is done by the cheapest conventional method and the union has great mechanical strength.

Sheet steel has been attached to cast iron by instantaneous welding in which the mere skin fusion of the cast-iron surface has not disturbed the structure of the material or resulted in any accumulation of heat which might cause distortion. This, however, is a very special process not ordinarily applicable to miscellaneous welding problems. Indeed, as time goes on the development of special processes in welding is steadily encroaching on the casting field in either existing or new designs, but this paper has been confined largely to the solution of the more usual problems in which the designer is confronted by the necessity of choosing between a welding and a casting.

Hydraulic-Laboratory Projects of the Corps of Engineers, U. S. Army

By LIEUT. F. H. FALKNER,[1] VICKSBURG, MISS.

Probably no other single engineering unit has had the privilege in recent years of undertaking a program of experimental hydraulics as broad in scope and as extensive in application as that carried on by the Engineer Department of the U. S. Army. The breadth of the program, involving 303 separate experiments conducted by 41 division and district offices scattered over the entire country, makes it impossible to deal with specific cases in this paper.

Hence, the purpose of this paper will be to present, in broad general outlines, the hydraulic-laboratory work of the Corps of Engineers. Specifically, it will touch upon the types of field problems subjected to experimental analysis, the facilities maintained for this work, and the accomplishments of the program. No effort will be made to explain the laboratory methods employed, since they conform generally with the methods and technique described by L. J. Hooper.[2] It is hoped that this paper will serve the engineering profession as an index of the facilities and sources of experimental data available in the Engineer Department of the U. S. Army.

INTRODUCTION

SINCE various nations have followed different courses in the assignment of engineering functions to departments, it may be of benefit to visiting engineers to explain the position of the War Department in American engineering. When our country first achieved its independence one of the first problems confronting its leaders was that of national defense and, in particular, defense of our coast line. This task naturally fell to the War Department which established a school for engineers and artillerists at West Point, N. Y., in 1803. The school later became known as the United States Military Academy. From this institution came the first organized group of American engineers and it fell to their lot to design and construct our first harbor defenses.

It was but a short step to add to the duties of this group the task of river and harbor improvements. Since in the early nineteenth century the widely scattered towns and settlements of this country had only the crudest overland avenues of communication, commercial intercourse depended primarily on water-borne transportation. The maintenance of these arteries, therefore, was of the greatest importance. Nothing could have been more natural than that the development and maintenance of these vital waterways should be entrusted to the only well-organized group of engineers in the government service. The work of administering our rivers and harbors has remained a function of the War Department since that time.

Today the War Department through its Corps of Engineers and the Engineer Department, consisting of more than 600 officers of the military service and more than 45,000 civilian employees, is charged with the improvement works on all our waterways for the benefit of navigation. Flood control, channel stabilization, harbor improvement, beach protection works, and related subjects of national character also come within the sphere of activity of the Engineer Department.

In 1838 the War Department undertook the first improvements on the Mississippi river when it was apparent that a sand bar threatened to destroy the harbor of St. Louis. A young Lieutenant of Engineers by the name of Robert E. Lee, who was destined to become the illustrious commander of the Confederate forces in the Civil war, was ordered to this assignment. The science of open-river regulation had not advanced far in that day, but the records show that Lieutenant Lee assiduously applied himself to a study of the river currents and field conditions. In spite of a paucity of stage and discharge data, his work was well done. The regulating works designed and constructed by Lee caused the sand bar to move downstream and provided deep water to St. Louis harbor. The harbor has remained open to this day due to the intelligent and scientific study of the problems by this young engineer officer.

From these early efforts we come to the beginning of the twentieth century and find that a more scientific approach to hydraulic problems is under way. The year 1903 saw the completion of Captain Gaillard's notable experiments at Duluth on the force of waves,[3] with which all engineers who have designed breakwaters are familiar. In 1910 experimental hydraulics again proved its value in the design of the locks for the Panama Canal. The next notable use of laboratory methods was in 1921 when the Engineer Department engaged the services of Professor W. C. Sadler, of the University of Michigan for extensive model tests to determine the proper bow-shape for river boats.[4]

It was not until 1929, however, that laboratory methods came into general use throughout the Engineer Department. With the establishment of the U. S. Waterways Experiment Station at Vicksburg, Miss., and the hydraulic laboratory of the U. S.

[1] Director, U. S. Waterways Experiment Station, Vicksburg, Miss. Lieutenant Falkner was graduated from the U. S. Military Academy in 1928 and served with the Third Engineers, Scofield barracks, Hawaii, until November, 1930, and then with the Sixth Engineers, Fort Lewis, Washington, from November, 1930, to August, 1931. He entered the University of California in August, 1931, as a graduate student in hydraulics, and in May, 1932, became assistant to the director of the U. S. Waterways Experiment Station, Vicksburg, Miss. In August, 1932, he entered the student-engineer school at Fort Belvoir, Va., and in May, 1933, became assistant director of the U. S. Waterways Experiment Station, Vicksburg, Miss. In July, 1934, he was appointed to his present position.

[2] "American Hydraulic Laboratory Practice," by L. J. Hooper, Trans. A.S.M.E., vol. 58, October, 1936, paper HYD-58-3, p. 577.

Contributed by the Hydraulic Division and presented at a meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, held at Niagara Falls, N. Y., September 17–19, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

[3] Wave Action in Relation to Engineering Structures. The Engineer School, Fort Belvoir, Va. Printed originally in 1904, reprinted in 1935.

[4] Preliminary report H. D. 857, 63rd Cong. 2nd Session; 1st Supplemental Report in H. D. 108, 67th Cong. 1st Session and Report of Chief of Engineers, 1922; 2nd Supplemental Report (not printed) June 17, 1929.

Engineer suboffice of the St. Paul District at Iowa City, Iowa, experimental hydraulics took a great stride forward and became an integral part of engineering design methods on all large projects. Within the past seven years the various hydraulic laboratories of the Corps of Engineers have undertaken 303 laboratory projects covering a wide field of problems.

CLASSIFICATION OF PROBLEMS STUDIED

The handling of this volume of experimental work has required the establishment of large laboratory facilities, but before describing the laboratories in detail, it would be well to examine the type of work required of them. A classification of the problems submitted for laboratory analysis will illustrate the broad scope of the work of the Corps of Engineers and explain why its laboratory facilities have been of varying design and at widely separated localities.

TABLE 1 CLASSIFICATION OF FIELD PROBLEMS AND EXTENT OF PARTICIPATION BY LABORATORIES

Group no.	Designation	—Number of investigations— U. S. Waterways Experiment Sta.	University labs.	Other govt. labs.	Total
1	Hydraulic features of fixed dams....	15	17	37	69
2	River canalization................	4	4	42	50
3	Open-river regulation for navigation	44	1	0	45
4	Flood control.....................	38	0	0	38
5	Coastal harbors and beaches.......	12	0	22	34
6	Miscellaneous projects............	51	7	9	67
	Totals..........................	164	29	110	303

TABLE 2 CLASSIFICATION OF LABORATORY PROJECTS ACCORDING TO TYPES OF PROBLEMS STUDIED

1 Hydraulic features of fixed dams
- (*A*) Overfall structures (spillways, stilling basins, bridge piers, aprons, baffles, fishways)
- (*B*) Pressure conduits and power equipment (outlet works, tunnels)
- (*C*) Seepage studies
- (*D*) Foundations

2 River canalization
- (*A*) Lock design
- (*B*) Movable-dam design
- (*C*) Effect of dam on flow conditions (flood backwater and low-water navigation)
- (*D*) Effect of dams on channel configurations (alignment of channel, silting)

3 Open-river regulation for navigation
- (*A*) By submerged sills
- (*B*) By wing dikes (permeable and impermeable groins)
- (*C*) By longitudinal dikes
- (*D*) By bank stabilization
- (*E*) By dredging

4 Flood control
- (*A*) By levees and levee realignment
- (*B*) By channel straightening
- (*C*) By diversions and outlets

5 Coastal harbors and beaches
- (*A*) Improvement of tidal estuaries and harbors for navigation
- (*B*) Breakwaters for reduction of wave action
- (*C*) Inlets, sandy coasts, and beaches

6 Miscellaneous projects
- (*A*) Improvement of model methods
- (*B*) Floating-plant designs
- (*C*) Wave studies
- (*D*) Stream-load studies
- (*E*) Protection of stream banks
- (*F*) Earth-embankment studies
- (*G*) Flow in closed conduits (general)
- (*H*) Fluid mechanics
- (*I*) Canal studies

Laboratory problems may be classified by two general methods: First, according to the method of laboratory treatment; and second, according to the type of field problem encountered. The first method of classification has been discussed in much detail so will not be repeated here. The second method of classification will be followed in this paper, primarily for the purpose of showing the types of problems with which the Corps of Engineers is principally concerned.

The problems subjected to laboratory treatment by the Corps of Engineers may be broadly grouped under the six general headings listed in Table 1. European practice generally provides a separate classification for canals. The number of such investigations by the War Department, however, does not justify this classification in this paper and these problems are grouped in Table 1 under miscellaneous projects. A better representation of the various laboratory projects can be obtained by subdividing each of the general classifications as shown in Table 2.

TABLE 3 SOILS LABORATORIES[a]

Laboratory (Engineers district or division)	Location	Date estab.	Type of projects investigated
U. S. Waterways Experiment Station (soils lab.) (L. M. V. D.)[b]	Vicksburg, Miss.	1931	Seepage studies. Standard soil analyses
Soils laboratory (Vicksburg Dist.)	Vicksburg, Miss.	1931	Seepage studies. Standard soils analyses
Fort Peck laboratory	Fort Peck, Mont.	1933	Earth-dam model cross-sections, and a model dam. Gelatin models to study stress patterns. Standard soils analyses
Soils laboratory (Zanesville Dist.)	Zanesville, Ohio	1934	Seepage studies of dams and study of possibility of piping in embankment of foundations of dams. Seepage studies of levees. Standard soils analyses
Soils-mechanics lab. (Memphis Dist.)	Memphis, Tenn.	1935	Subsurface studies for levees and earth-fill dams. Standard soils analyses
Soils laboratory (Eastport Dist.)	Eastport, Me.	1935	Seepage studies. Stress and settlement studies of rock-fill dams on plastic foundations. Standard soils analyses
Soils-mechanics lab. (Tucumcari Dist.)	Conchas dam, N. M.	1936	Model tests of dike sections. Standard soils analyses
Denison-soils lab.	Denison, Texas	1936	Seepage studies. Standard soils analyses

[a] Listed in order of date of establishment.
[b] Lower Mississippi Valley Division.

TABLE 4 TEMPORARY HYDRAULIC LABORATORIES[a]

Laboratory (Engineer district or division)	Location	When used	—Projects investigated— Type	No.
Fort Mifflin (Philadelphia Dist.)	Fort Mifflin, Pa.	1921 and 1928	Tidal models to study channel improvements for navigation	2
Bonnet Carre spillway hydraulic lab. (2nd New Orleans dist.)	At site of Bonnet Carre spillway	1928	Spillway model	1
1st New Orleans Dist.	Burwood, La.	1930	River model for channel improvement (Head of Passes)	1
2nd New Orleans Dist.	Government fleet, New Orleans, La.	1932	Towboat stability tests	1
Gasconade boatyard (Missouri River Division)	Government boatyard, Gasconade, Mo.	1934	Spillway models (for Fort Peck dam)	2
Dry dock (Louisville Dist.)	Louisville and Portland Canal, Louisville, Ky.	1934	Model tests to determine a method of eliminating scour below bear-trap section of dams	1
Hydraulic laboratory (Missouri River Division)	Postal Telegraph bldg., Kansas City, Mo.	1935–1936	Models of tunnel and control towers, and of tunnel outlet works (for Fort Peck dam)	2
Mobile Dist.	Spring Hill, Ala.	1935	Investigation of cross currents in channels	1

[a] Listed in order of date of establishment.

The classification in Table 2 shows the extent of the work undertaken in the laboratories of the Corps of Engineers, and reveals the reasons for utilizing existing university laboratories. A complete list of all these projects will be found in the Appendix. That these problems are encountered on waterways ranging from the Atlantic to the Pacific Oceans and from Canada to Mexico further explains the division of laboratory effort among widely scattered institutions. The emergency work of the past few years frequently has required laboratory facilities near the site of the work. The services of existing laboratories have been

TABLE 5 PERMANENT HYDRAULIC LABORATORIES[a]

Laboratory (Engineer district or division)	Location	Date estab.	Capacity, no. projects	Projects investigated: Type	Projects investigated: No.
U. S. Waterways Experiment Station (L. M. V. D.)	Vicksburg, Miss.	1929	30	Flood-control studies open-river navigation projects, canalization projects, hydraulic features of fixed dams, tidal and wave-action studies for harbors and inlets	181
U. S. Engr. Suboffice (St. Paul District)	Hydraulic Laboratory, Iowa City, Iowa	1929	12	Fixed and movable-bed river models, fixed spillways, movable-crest spillways, percolation rates and paths, lock hydraulic systems, silt movement, pile foundations	46
Caisson plant (Milwaukee District)	Milwaukee harbor, Wis., off McKinley beach	1931	1	Various types of breakwater construction and wave action	4
U. S. Beach-Erosion Board wave tank	Ft. Belvoir, Va.	1932	1	General investigation of the movement of sand under the action of tides and waves, general study of waves, study of the effect of protective works on sand movement	9
Linnton Hydraulic Laboratory (2nd Portland Dist.)	Government moorings, Portland, Oreg.	1934	3	Investigations by means of models of river, spillway, and other details of Bonneville dam, to determine best designs	5 principal models have been built. All pertain to Bonneville dam project
U. S. Tidal Model Laboratory (N. Pacific Division)	Univ. of California, Berkeley, Calif.	1934	2	Bed-load movement, effect of existing and proposed structures in tidal channels, and energy required for pumping various sands	9

[a] Listed in order of date of establishment.

engaged or new laboratories been established as the occasion demanded. Because of the diverse character of the problems encountered and the frequent need for proximity of facilities, the nature of those facilities will be discussed.

INDEPENDENT UNIVERSITY LABORATORIES

It may have been noted in Table 1 that a total of 29 laboratory investigations have been submitted to university laboratories. These tests have been conducted at the Massachusetts Institute of Technology by Prof. K. C. Reynolds, at the Alden hydraulic laboratory of Worcester Polytechnic Institute by Prof. C. M. Allen, at the Carnegie Institute of Technology by Prof. H. A. Thomas, at the Case School of Applied Science by Prof. G. E. Barnes, and at the University of Michigan by Prof. H. W. King. The ability of their personnel and their proximity to important projects of the Corps of Engineers are primarily responsible for the selection of these laboratories. These laboratories are entirely independent of the Corps of Engineers and will not be described.

LABORATORIES OF THE CORPS OF ENGINEERS

Of the 14 hydraulic laboratories established by the Corps of Engineers only six may be designated as permanent units. In addition to these distinctly hydraulic laboratories, eight soils mechanics laboratories, the functions of which are closely related to hydraulic investigations, have been established. Table 3 lists the soils laboratories, while Tables 4 and 5 distinguish between temporary and permanent hydraulic laboratories.

Soils Laboratories

While a large portion of the work of most of the soils laboratories listed in Table 3 is not related directly to hydraulic-engineering work, all of these laboratories are engaged in seepage studies which fall in the classification of "hydraulic features of fixed dams." The equipment of these laboratories is fairly well standardized and consists of long, narrow flumes for the study of seepage lines through earth models of levees and dams, in addition to the usual equipment for soils-mechanics tests. The Fort Peck and Zanesville laboratories are unique in the use of gelatin models and photoelastic methods for the study of stress patterns. Fig. 1 shows one of the test flumes at the Zanesville laboratory. Fig. 2 shows the interior of the Memphis soils laboratory. Fig. 3 shows test dam No. 1 built by the soils laboratory at Tucumcari, N. M. Figs. 4 and 5 are views of the soils laboratory at the Passamaquoddy tidal-water project in Maine.

Temporary Hydraulic Laboratories

Each of the eight temporary laboratories listed in Table 4 was established for the study of a single problem. Three of these made use of a level outdoor space and a fourth was installed in the basement of a large office building. The other four made ingenious use of equipment designed for other purposes. The

Fig. 1 A Test Flume at the Zanesville Laboratory, Zanesville, Ohio

Fig. 2 The Soils Laboratory, Memphis, Tenn.

Fig. 3 Test Dam No. 1 Constructed by Soils Laboratory, Tucumcari, N. M.

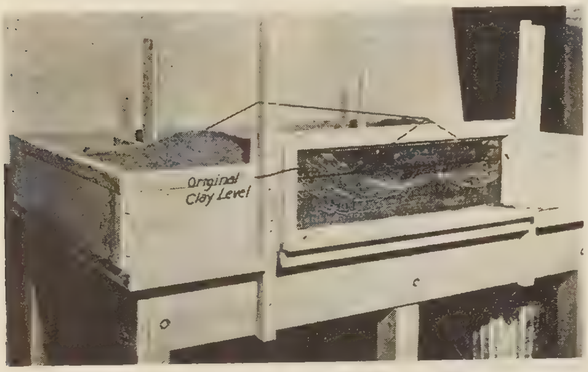

Fig. 4 Table for Model Study of Settlement Characteristics of Proposed Structures of Division No. 1 at the Passamaquoddy Tidal-Water Project in Maine

Fig. 5 Flume for Seepage Studies in the Soils Laboratory at the Passamaquoddy Tidal-Water Project in Maine

Louisville District installed a model in a dry dock not in use and took advantage of the dry-dock flooding system for water supply. New Orleans built its tow-boat models on an idle river barge. Mobile constructed its model directly in the bed of a natural stream and diverted the water supply through the structure. The most elaborate arrangement of all was used by the Missouri River Division for its model of the Fort Peck spillway. Since the model required up to 82 cfs of water, a 36-in. dredge was tied up to the bank to supply water directly from the Missouri River. The huge spillway model and its unusual water-supply system are shown in Fig. 6.

Permanent Hydraulic Laboratories

U. S. Tidal-Model Laboratory, Berkeley, Calif. This laboratory, established in 1934 at the University of California by the North Pacific Division, is shown in Fig. 7 and consists essentially of a model basin 38 by 58 ft in plan with an elaborate piping system permitting water supply from all sides of the basin. Prof. M. P. O'Brien, consulting engineer in charge, has developed a very accurate tide mechanism consisting of rotating concentric cylinders with variable port openings for regulating the rate of

Fig. 6 Fort Peck Spillway Model, Gasconade, Mo.

Fig. 7 U. S. Tidal Model Laboratory, Berkeley, Calif.

Fig. 8 Cofferdam Model for Bonneville Dam at the Linnton Hydraulic Laboratory

water supply according to the time scale of the tidal cycle. The addition of a flume 3 ft in width permits this laboratory to carry on two projects concurrently. To date it has undertaken nine investigations. The laboratory is designed primarily to study currents and sand movements in tidal estuaries.

Linnton Hydraulic Laboratory, Portland, Oreg. This laboratory, established in 1934 by the Second Portland District, has constructed five principal models. All models pertain to the Bonneville dam project; one of these is shown in Fig. 8. The laboratory can carry on three projects simultaneously, and is concerned with problems of river canalization, cofferdam design, spillway design, and related subjects.

U. S. Beach-Erosion Board Wave Tank. A 16 × 22-ft wave tank is housed at Fort Belvoir, Va. A permanent staff of three engineers carries on a continuous program of general research on the movement of sand under the action of tides and waves. The main features of the program of this laboratory are fundamental research on wave forms; effect of height, frequency, and length of waves on sand beaches; and the action of inlets on sandy coasts. Nine general studies have been completed since the start of work in 1932.

Caisson Plant, Milwaukee Harbor, Wis. In 1931 the district engineer at Milwaukee established this laboratory for the purpose of continuing the work which Captain Gaillard discontinued in 1903. A large wave tank 40 ft long, 6 ft deep, and 8 ft wide has been used for four projects thus far. The laboratory can handle but one project at a time and has directed its program toward the solution of breakwater design, research on wave pressures, and the dissipation of wave energy. Laboratory work has been supplemented by full-scale field measurements on the north breakwater at Milwaukee.

Hydraulic Laboratory of the U. S. Engineer Suboffice, St. Paul, District. This laboratory, the facilities of which are owned by the Iowa Institute of Hydraulic Research and are used on a rental basis by the St. Paul District, is located at Iowa City, Iowa, on the bank of the Iowa River just downstream from a low-head dam.

The lake created by the dam supplies water to a 10 × 10-ft concrete channel approximately 300 ft long which extends from the end of the dam to and beneath the laboratory building. It also supplies a second channel 16 ft wide located beneath the building and connected to the intake at the dam by a 48-in. steel pipe.

The laboratory building, shown in Fig. 9, has a central tower section which is approximately 45 ft square and five stories high. The two end wings are approximately 25 ft and 30 ft wide, respectively and 60 ft long. Beneath the first floor and above the concrete channels are located pump rooms, carpenter shop, tool room, and volumetric-measuring basins. City water is used in the circulating systems. Provision has been made for approximately 25 cfs pumping capacity.

The first two floors of the building are devoted entirely to model testing. The fixed equipment on these floors includes two glass-sided flumes, two 20,000-lb weighing tanks, two shallow steel tanks, and one orifice tank for classroom instruction. The third floor contains classrooms, storerooms, machine and carpenter shops, and two constant-head tanks. The fourth and fifth floors contain offices, a library, and a drafting room. An annex to the laboratory provides space for large river-channel models.

These facilities are all available to the Engineer Department and the work carried on here is greatly facilitated by consultation with the able staff of the Iowa Institute of Hydraulic Research.

U. S. Waterways Experiment Station, Vicksburg, Miss. The U. S. Waterways Experiment Station is located on a federal reservation containing 245 acres. Two artificial lakes constitute the source of water supply for most of the studies. The principal building at the station contains a large experiment hall, offices, and a photographic laboratory. Auxiliary buildings contain a soils laboratory, various shop facilities, and warehouses. A general view of the station is shown in Fig. 10.

Fig. 9 Iowa Institute of Hydraulic Research, and Location of Hydraulic Laboratory of the U. S. Engineer Suboffice of the St. Paul District

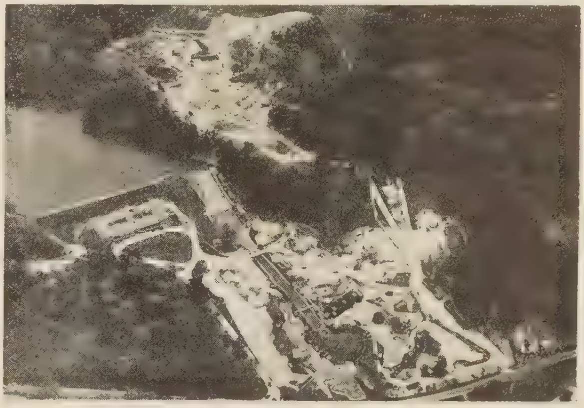

Fig. 10 U. S. Waterways Experiment Station, Vicksburg, Miss.

Fig. 11 Pumping Unit No. 5 at the U. S. Waterways Experiment Station

Model studies are conducted both inside the experiment hall and on the 35 acres of outdoor experiment fields. Some of the latter models have temporary shelters.

Fixed equipment inside the experiment hall includes volumetric measuring tanks, velocity-meter calibrating equipment, and four flumes of the following dimensions: A steel flume 3.5 × 3 × 165

ft with glass side panels; a tilting glass-sided flume 1 × 2 × 26 ft; a tilting concrete-lined flume 2.3 × 1.3 × 48 ft; and a wooden flume 8.5 × 6 × 58 ft. Each flume is equipped with a measuring weir, and is supplied with water from pumping units. The large wooden flume is well suited for studies of spillways and stilling basins.

Water from the larger lake is supplied from two 20-in. gravity ines, each of 20 cfs capacity, and a 7-ft gravity line, the capacity of which for short infrequent periods is 700 cfs. The supply from the lake is supplemented by five pumping units, the capacities of which are 7, 7, 10, 31, and 15 cfs, respectively, or a total of 70 cfs. The smaller lake is used as a reservoir for one of the pumping units, and is supplied from the large lake by a booster unit. Fig. 11 shows the 15-cfs pumping unit.

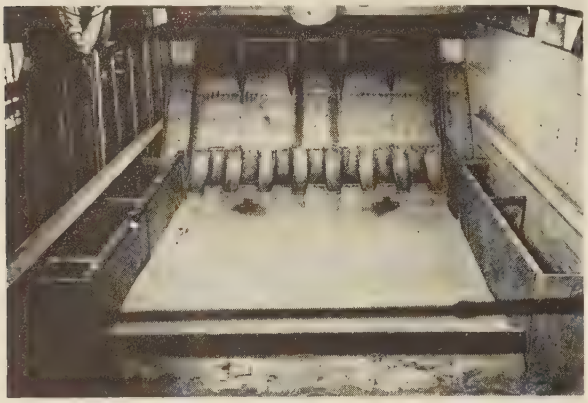

Fig. 12 Conchas Dam Model Showing Dentated Bucket and Subdam Preliminary Design

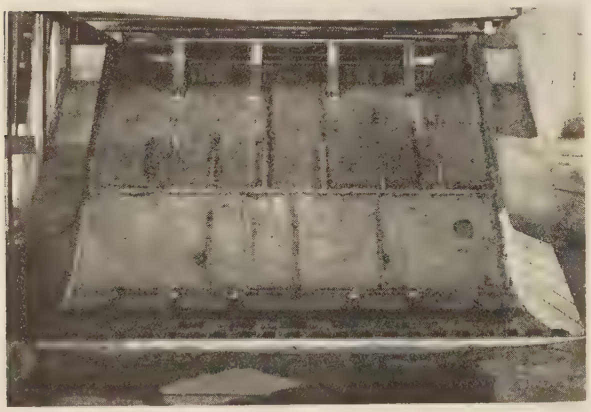

Fig. 13 Conchas Dam Model Showing Final Design With Horizontal Apron, Baffles, and End Sill

The personnel of the station varies from 200 to 400 persons depending on the volume of work. Of this number, about half are in the experiment section and soils laboratory, and the remainder are in the administration and general service sections.

The personnel and equipment at the station are sufficient to carry on simultaneously from 20 to 25 active hydraulic model studies. In addition, the soils laboratory can conduct simultaneously several problems connected with soil mechanics.

Results of Model Tests

The results of the experimental program conducted by these laboratories can be best illustrated by the selection of a typical problem from each of the six groups of the general classification. In the selection of these six examples the author has been prompted to take all of them from the work of the U. S. Waterways Experiment Station in order to avoid discussing problems executed under another's direction. The Vicksburg institution has taken the lead in the field of flood control and open-river regulations and has a preponderance of projects in the miscellaneous classification, whereas other government laboratories have undertaken more work in the fields of river canalization, hydraulic features of fixed dams, and coastal harbors and beaches. In these latter groups there are numerous experiments equally if not more noteworthy than the ones selected from the Vicksburg experience. An Appendix listing all laboratory projects of the Corps of Engineers together with the name of the laboratory and the office requesting the study is given at the end of this paper. Examination of this Appendix will show the extent of participation of each laboratory in any particular field of investigation.

Hydraulic Features of Fixed Dams

Conchas Dam Stilling-Basin Model. The Conchas dam stilling-basin model was constructed in order to check a proposed design

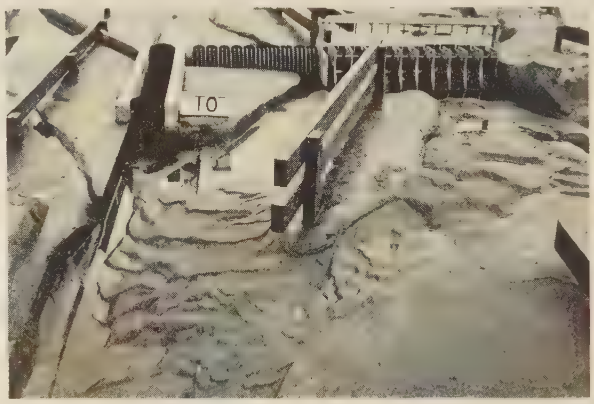

Fig. 14 Upstream View of Starved Rock Lock and Dam, Illinois River. Models With Cofferdam Removed Showing Shoaling in Entrance to Lock

for destroying the excess energy at the toe of the 215-ft high overflow spillway of Conchas dam, near Tucumcari, N. M.

The features of this design shown in Fig. 12 were almost identical to that of the Tygart dam stilling basin, since the hydraulic characteristics of the Conchas dam are similar with one exception —height of tailwater. There was every reasonable expectation that the Tygart plan of a dentated drop-off bucket and subdam would be applicable.

The model indicated, however, that the design was not suited to the peculiarities of the site. A total of 294 tests on several different stilling-basin designs prepared by the Tucumcari District led to the adoption of an entirely different type of stilling basin, which is shown in Fig. 13. This modification resulted in a reduction of the estimated cost by about $625,000.

River Canalization

The Starved Rock Lock and Dam, Illinois River. The Starved Rock lock and dam, shown in Fig. 14, located approximately 1 mile above Utica, Ill., forms an integral part of the plan for improving the Illinois River. After the construction of the structure was completed, the channel below the downstream end of the lock shoaled. Dredging was required at intervals to maintain project dimensions.

In 1932, authority was granted for a model study of this problem. It was desired, in general, to determine (*a*) the cause of the shoaling and the means of preventing same; (*b*) whether an existing cofferdam wall, located below the lock should be removed or retained, also whether a more permeable structure should be constructed to replace the cofferdam; (*c*) the most suitable method of operating the Taintor gates; (*d*) successive shore lines, and areas of shoaling, when the two islands immediately below the dam were allowed to erode.

The model tests indicated that the shoaling between the cofferdam and the guide wall could be prevented by closing a breach at the upper end of the cofferdam. This closure was effected in the prototype and no shoaling has occurred there for the past four years.

Shoaling below the cofferdam was found to be caused principally by material eroded from the sides of two islands below the dam. The model indicated that this erosion would gradually diminish and the shoaling abate if the Taintor gates were all opened equal amounts. Tests also showed that the most undesirable method of operating the Taintor gates was to pass the

Fig. 15 Upstream View of St. Clair River Model

entire discharge through the two gates nearest the lock. Under this condition the model indicated that a half of Leopold Island would be washed away and excessive shoaling of the navigation channel might be expected. Unfortunately during the winter of 1934–1935 all Taintor gates were frozen solid except the two nearest the lock. Flood waters required that they be opened and surveys of February and March, 1935, disclosed how deadly accurate the model had been. Up to that time negligible shoaling had occurred, but the 1935 surveys showed half of Leopold Island gone and the channel badly shoaled. The model had pointed the way, but nature prevented proper execution of the plan. The situation has since been remedied.

Fig. 16 Aerial View of Model of Lower Mississippi Valley

Open River Regulation

Improvement of the St. Clair River by Sills. For some years past, there has been a slight but perceptible lowering of the levels of Lakes Michigan and Huron. Since one of the principal factors limiting present navigation on the Great Lakes is the depth of harbors, this decrease in lake level is a cause of much concern to the shipping interests of this region. Therefore, two models of the St. Clair River were constructed to determine the backwater effects of submerged sills in the river as a means of increasing depths in Lake Huron. The combination of number, type, and location of sills necessary to produce 0.54 ft backwater was to be determined along with the effects of the sills on the distribution of velocities in the river channel. Tests were first made in a steel flume to determine comparative effects from sills of different cross-sections. A fixed-bed model with a vertical scale of 1:30 and a horizontal scale of 1:100 was next used for investigations, after which a second model, undistorted, to a scale of 1:100 was constructed. This model is shown in Fig. 15. The indications from the investigations were that it is possible to secure the desired rise in the level of Lake Huron by use of sills at the proposed locations, but that it is necessary to use sills of different cross sections from those proposed in order to obtain the desired effects, and that alternative locations for sills in the same reach of the river may be used.

The most important feature of this investigation was the

experimental data concerning the shape of the sills. By varying the cross section of a sill it was found possible to increase or decrease its efficiency in creating backwater by as much as 100 per cent. The slope of the upstream face was found most important while the downstream slope appeared to have negligible effect within a limited range. These studies resulted in a radical change of design for the sills.

FIG. 17 LOOKING SOUTH FROM VICKSBURG ON THE MISSISSIPPI VALLEY MODEL SHOWING THE MAIN CHANNEL, RED RIVER BACKWATER, AND THE ATCHAFALAYA BASIN

FIG. 18 MODEL OF PORT WASHINGTON, WIS.

FLOOD CONTROL

Mississippi River, Helena, Ark. to Donaldsonville, La. The model constructed to study the general features of the flood-control plans for the lower Mississippi Valley is by far the most ambitious laboratory investigation of the Corps of Engineers. The model, shown in Fig. 16, includes the entire overflow area of the alluvial plain of the Mississippi south of Helena, Ark. It includes 602 miles of the main river, its five principal tributaries, all backwater areas, and the entire Atchafalaya Basin to the Gulf of Mexico; a total area of 16,000 sq miles. The model itself is 1100 ft long. A close-up of part of the structure is shown in Fig. 17.

In operation, 42 engineers attend the 17 water-supply lines and read the 210 gages. Flood years are represented on time schedule, the daily changes in discharge of each stream being made by the operators, and river gages being read daily. The form, height, and time of travel of the flood waves are recorded, and the routing of the flood waters through the intricate system of channels and reservoirs is carefully checked. Three known floods of varying magnitude have been reproduced faithfully and the accuracy of the model has been established.

Four studies have been undertaken in the past year. These involved (1) determining the efficiency of the cutoffs as they existed in 1935, (2) tracing the change in channel capacity in a restricted reach, (3) studies of the effect of future cutoff developments in the main channel, and (4) flood routing through the Atchafalaya Basin with various designs of diversion outlets. The studies at present are for the purpose of determining the extent and sequence of work required for handling the 1927 flood and the project superflood.

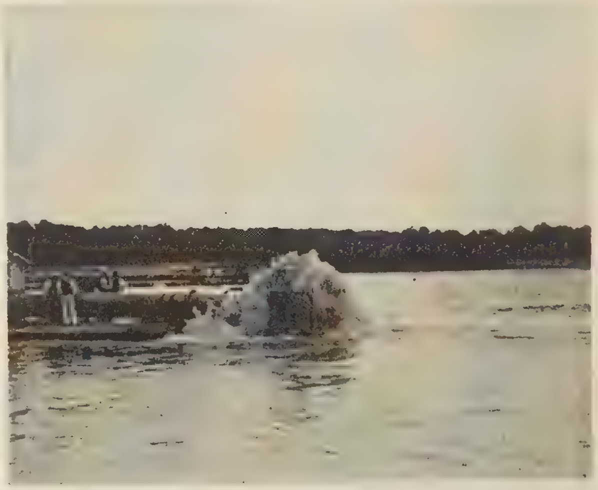

FIG. 19 DISCHARGE FROM DREDGE JADWIN THROUGH 32-IN. PLAIN PIPE AT A PUMP SPEED OF 150 RPM. MEASUREMENTS SHOWED 13.3 PER CENT SOLIDS

FIG. 20 DISCHARGE FROM DREDGE JADWIN THROUGH A 32-IN. TYPE-12 RIFFLED DISCHARGE PIPE AT A PUMP SPEED OF 150 RPM. MEASUREMENTS SHOWED 18.1 PER CENT SOLIDS

COASTAL HARBORS

The Port Washington Model. In December, 1934, the construction of a new north breakwater was completed in Lake Michigan at Port Washington Harbor, Wis. Subsequent storms indicated that the breakwater was not effective in reducing

the wave heights within the basin and slips, and it was proposed to build an extension to the south breakwater at a cost of about $200,000, in an effort to reduce the wave heights within the harbor.

Before beginning construction of the proposed breakwater, the problem was submitted to the U. S. Waterways Experiment Station for investigation by means of a hydraulic model. The specific purpose of the model study was to determine the degree of closure of the entrance necessary in order to effectively still the harbor. The district engineer defined 300 ft as being the minimum allowable width at the entrance.

The model shown in Fig. 18 was constructed to an undistorted scale of 1:50 and a wave machine was constructed which could be shifted to produce waves from several different directions. The results of the model study showed that the construction of the proposed south breakwater would not noticeably reduce the waves when subjected to easterly storms. A more efficacious construction was found to be the placing of riprap cribs at points of wave reflections in the inner harbor; these cribs absorbed the energy in the wave forms and greatly reduced their height.

Miscellaneous Projects

Experiment With Pipe-Line Mixers. An outstanding study among the various investigations which have been made that fail to fall under any general classification is the experiment with pipe-line mixers. The problem deals with the transportation of solids through the discharge lines of hydraulic dredges and a method to increase the percentage of solids that can be passed for the same power. While the idea of mixing the material to prevent all the solids from moving on the bottom is not new, having been tried by the Memphis Engineer District with a diametrical warped plate in 1912, this is the first instance that rifles have been used for mixing. The present study is only for dredges pumping sand. The apparatus for the investigations consists of a circulating system in which the per cent of solids can be varied at will and a 4-in. test line in which the different designs of mixers are installed. The investigations deal with height, length, pitch, and spacing of the rifles in the line. At the present time 20 different designs have been investigated at the laboratory and one of these has been tested in the field. Although somewhat impractical for field installation, the mixer tested by the Memphis Engineer District early in June, 1936, showed an increase in material carried of 45 per cent over plain pipe. The present aim of the study, now that it has been found that the mixers will prove satisfactory, is to develop a section that can be easily handled in the field. Field and laboratory tests are being energetically pursued and it is expected that dredging operations in 1937 will exhibit a marked increase in efficiency. Figs. 19 and 20 illustrate what has already been accomplished.

GENERAL CONCLUSIONS

Since the laboratory work of the Corps of Engineers has been confined to the investigation of particular structures of waterways the results of the program cannot readily be analyzed with a view to correcting or improving the fundamental concepts of hydraulics. Although few new theories of hydraulics have been brought forth, and the basic laws of hydraulics and of hydraulic similitude have not been greatly clarified by this experimental program, the work has added a considerable quantity of experimental data to our fund of knowledge. The experiments have covered a great number of specific problems which may again arise, and a review of the experiments in a particular field may supply needed data for future problems. During the past year the U. S. Waterways Experiment Station has had several cases in which data from previous tests could be applied. In these cases recourse to model experiments was unnecessary and the cost thereof was eliminated. While this has been accomplished in a few cases, it is obvious that the 303 investigations do not begin to cover all the special cases which nature presents to the hydraulic engineer. Many of these investigations are extremely limited in application and have only partially solved the particular phenomenon under study. As long as our mathematical analyses of hydraulic problems (and in particular of such phenomena as bed-load movement, sedimentation, causes of crosscurrents, turbulences, and backwater) remain as inaccurate and incomplete as they are today, recourse to hydraulic model experimentation will be essential to hydraulic engineers.

It was stated that no general research work was done by the War Department. There are two exceptions to this statement. The personnel of the beach-erosion board wave tank have spent four years studying wave forms and the effects of tides and waves upon sandy beaches. The program is progressing rapidly and although knowledge of beach actions is being gained, proper substantiation of their findings is yet needed before publication of the results. The second exception is the program of experiments undertaken at the U. S. Waterways Experiment Station for the improvement of model methods and technique. The outstanding achievements of this latter program may be grouped as follows:

(*a*) The application of low-specific-gravity materials to use as bed load and sediment in movable-bed models. This development has broadened the field of application of model methods.

(*b*) A better understanding of the limitations of geometric distortion in models and a resultant increase in accuracy of model design methods.

(*c*) The improvement of equipment for generating tides and waves and for measuring velocities. (This work is being carried on at all permanent laboratories of the Corps of Engineers.)

The experience of the past seven years points to the following general conclusions regarding the laboratory projects of the Corps of Engineers:

1 Having undertaken over 300 laboratory investigations, the Corps of Engineers possesses a large fund of experimental data which may be applied to future problems.

2 The application of experimental hydraulics will be essential to hydraulic problems until such time as a more complete mathematical treatment of hydraulics is developed.

3 The reliability of a model study will be in direct proportion to (*a*) the accuracy of the field data, (*b*) the ability of the experimenter and his knowledge of model limitations, and (*c*) the degree of coordination between field and laboratory and the knowledge of field and laboratory engineers concerning prototype phenomena.

4 Research work is continually improving model methods and technique and has already improved model accuracy and broadened the scope of model application.

5 Hydraulic model studies almost without exception pay large dividends by warning against unsafe practices or by pointing the way to improved designs.

6 No important improvement of waterways should ever be constructed without at least a thorough knowledge of laboratory experiments in the particular field, and in most cases model experimentation will be necessary.

Appendix[5]

1 Hydraulic Features of Fixed Dams

(A) Overfall structures (spillways, stilling basins, bridge-piers, aprons, baffles, fishways)

Name of study	Office authorizing study	Where conducted
Ogee spillway tests[6]	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Sand dams[6]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Mississippi lock and dam No. 7 (Onalaska spillway)[7]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Spillway model Bonneville dam[6]	2nd Portland Dist.	Linnton hydraulic laboratory
Diffusing chamber Bonneville dam[6]	2nd Portland Dist.	Linnton hydraulic laboratory
Mill Creek diversion dam[6]	2nd Portland Dist.	Linnton hydraulic laboratory
Bonnet Carre spillway	2nd New Orleans Dist.	Laboratory located at site of spillway
Tygart River dam[6]	Pittsburgh Dist.	Carnegie Tech.
Bluestone dam, New River	Huntington Dist.	Carnegie Tech.
Fort Peck spillway[6]	Board of Consulting Engineers for Fort Peck dam	Gov't Boatyard at Gasconade, Mo.
Outlet works and spillway of Dover dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works and spillway of Mohicanville dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works and stilling basin of Clendening dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Dover dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Mohawk dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Senecaville dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Piedmont dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Wills Creek dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Tappan dam[7]	Zanesville Dist.	Case School of Applied Science
Outlet works of Charles Mill dam[7]	Zanesville Dist.	Case School of Applied Science
Spillways St. Lucie Canal, Florida[6]	Gulf of Mex. Div.	U. S. Waterways Expt. Sta.
Spillways St. Lucie Canal, Florida, redesign[6]	Gulf of Mex. Div.	U. S. Waterways Expt. Sta.
Conchas dam stilling basin[6]	Tucumcari Dist.	U. S. Waterways Expt. Sta.
Brown Lake spillway	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Effects of overflow on railroad embankments[7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(B) Pressure conduits (outlet works, power equipment tunnels)

Name of study	Office authorizing study	Where conducted
Norris dam diversion tunnels, Clinch River Tenn.	Chattanooga Dist	U. S. Engr. suboffice, Iowa City, Iowa
Mississippi lock and dam No. 7 Onalaska spillway[7]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Filling gate tests, Passamaquoddy tidal-power project[6]	Eastport Dist.	Alden hydraulic lab.
Cavitation resistance tests, Passamaquoddy tidal-power project	Eastport Dist.	M.I.T.
Materials-corrosion investigation of metals in sea water for Passamaquoddy tidal-power project	Eastport Dist.	Materials-corrosion laboratory, Eastport, Me.
Model study for determination of casing and draft-tube design Passamaquoddy tidal-power project[6]	Eastport Dist.	At laboratories of four turbine manufacturers
Fort Peck diversion tunnels[6]	Missouri River Div.	Missouri River Div. lab.
Conduits for Madden dam	Office, Chief of Engrs.	Carnegie Tech.
Outlet works of Mohawk dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works and spillway of Dover dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works and stilling basin of Clendening dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Senecaville dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Bolivar dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Pleasant Hill dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works and stilling basin of the Piedmont dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works and spillway of the Mohicanville dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Wills Creek dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Tappan dam[6]	Zanesville Dist.	Case School of Applied Science
Outlet works of Charles Mill dam[6]	Zanesville Dist.	Case School of Applied Science

(C) Seepage Studies

Name of study	Office authorizing study	Where conducted
Sand-dike tests	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Weep holes	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Bolivar model study[6]	Zanesville Dist.	Zanesville lab.
Mohawk model study[6]	Zanesville Dist.	Zanesville lab.
Beach City model study[6]	Zanesville Dist.	Zanesville lab.
Atwood model study[6]	Zanesville Dist.	Zanesville lab.
Wills Creek model study[6]	Zanesville Dist.	Zanesville lab.
Mogadore model study	Zanesville Dist	Zanesville lab.
Cowan Creek model study	Zanesville Dist.	Zanesville lab.
Nimisila model study[6]	Zanesville Dist.	Zanesville lab.
Magnolia levee model study[6]	Zanesville Dist.	Zanesville lab.
Sandyville levee model study[6]	Zanesville Dist.	Zanesville lab.
8 Earth dam model sections	Ft. Peck Dist.	Fort Peck lab.
Foundation model tests Conchas dam	Tucumcari Dist.	Tucumcari lab.
Small-scale models of impervious core materials Conchas dam	Tucumcari Dist.	Tucumcari lab.
Model tests of design sections Conchas dam	Tucumcari Dist.	Tucumcari lab.
Sardis dam—2 models	Vicksburg Dist.	Vicksburg soils lab.
Efficacy of sublevees and berms for control of seepage[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Investigation and sug-	St. Louis Dist.	U. S. Waterways

[5] This tabulation of laboratory projects undertaken by the Corps of Engineers, U. S. Army, does not include three relief-map projects or 15 miscellaneous soil investigations of minor character. See Tables 1 and 2 for classification of projects and total number in each classification. Reports on many of these projects are available from the district authorizing the study. Inquiries should be addressed to the attention of the respective district engineers.

[6] Report on study is complete.

[7] Listed under more than one heading.

Name of study	Office authorizing study	Where conducted
gested design of material for a levee; Mauvais Terre levee and drainage district[7]		Expt. Sta.
Investigation of borrow-pit material for levee at Tulsa, Okla.	Memphis Dist.	U. S. Waterways Expt. Sta.
Complete design of supplementary dam and subsoil exploration at U. S. Waterways Experiment Station[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Estimate of seepage into reservoirs in White River backwater area[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Impervious core walls in levees, 1st study[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Impervious core walls in levees, 2nd study[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(*D*) Foundations

Name of study	Office authorizing study	Where conducted
Percolation through foundation materials	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Pile-foundation tests	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Sand consolidation study	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Weep holes	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Rock-fill-dam tests	Eastport Dist.	Alden hydraulic lab.
Bolivar model study[7]	Zanesville Dist.	Zanesville lab.
Mohawk model study[7]	Zanesville Dist.	Zanesville lab.
Beach City model study[7]	Zanesville Dist.	Zanesville lab.
Atwood model study[7]	Zanesville Dist.	Zanesville lab.
Wills Creek model study[7]	Zanesville Dist.	Zanesville lab.
Mogadore model study[7]	Zanesville Dist.	Zanesville lab.
Cowan Creek model study[7]	Zanesville Dist.	Zanesville lab.
Nimisila model study[7]	Zanesville Dist.	Zanesville lab.
Borrow-pit investigation Conchas dam	Tucumcari Dist.	Tucumcari lab.
Compaction tests	Tucumcari Dist.	Tucumcari lab.
Analysis of borrow material, Bear Canyon dam, N. M.	WPA State of New Mexico	Tucumcari lab.
Analysis of samples Pajarito Creek dam	Natl. Park service	Tucumcari lab.
Analysis of samples Miami dam, N. M.	WPA State of New Mexico	Tucumcari lab.
Investigation of material from nine proposed dam sites in Ark., Mo. and Okla.	Lower Miss. Valley Div.	Memphis Dist. soils lab.
Subsidence investigations[6]	Vicksburg Dist.	U. S. Waterways Expt. Sta.
Foundations and design for levee; Clear Creek levee and drainage district[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Investigation of embankment (levee) foundations for stability, applying Jurgenson's method of analysis[7]	Zanesville Dist.	U. S. Waterways Expt. Sta.
Soil and rock tests for Conchas dam[6]	Tucumcari Dist.	U. S. Waterways Expt. Sta.
Soil tests for Bear Canyon dam[6]	Tucumcari Dist.	U. S. Waterways Expt. Sta.

2 River Canalization

(*A*) Lock design

Name of study	Office authorizing study	Where conducted
Lock and dam No. 2, Kiskiminitas River, Pa.[7]	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Pickwick lock hydraulic system, Tenn. River	Nashville Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, Keokuk lock, Keokuk, Iowa[6]	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Tenn. River, Guntersville lock hydraulic system	Nashville Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Lock culverts	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Tenn. River, Chickamauga lock hydraulic system	Nashville Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Navigation lock tests	Eastport Dist.	Alden hydraulic lab.
Passamaquoddy tidal power project[6]		
Lock No. 3, Miss. River[6]	St. Paul Dist.	Zanesville lab.

(*B*) Movable-dam design

Name of study	Office authorizing study	Where conducted
Lock and dam No. 15, Miss. River, Rock Island, Ill.[7]	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Bear Trap dam No. 30, Ohio River[6]	Cincinnati Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Roller-gate pressure tests	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Submergible Taintor gate	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 22, Hannibal, Mo.	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Illinois River, Peoria and La Grange dams	Chicago Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Lock and dam No. 2, Miss. River, Hastings, Minn.[7]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Marmet lock and dam, Kanawha River, Marmet, W. Va.[6]	Huntington Dist.	U. S. Engr. suboffice Iowa City, Iowa
Ogee spillway tests	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Roller gate stilling basins	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Winfield lock and dam stilling basin Kanawha River, Winfield, W. Va.	Huntington Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 11, Dubuque, Iowa	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 20, Canton, Mo.[7]	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Ohio River, Montgomery Island lock and dam[7]	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Monongahela River, new lock and dam No. 4	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, lock and dam No. 20, Canton, Mo.	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Scour below Bear-traps	Louisville Dist.	Louisville and Portland canal dry dock
Bonneville spillway gates	2nd Portland Dist.	Linnton hyd. lab.

(*C*) Effect of dams on flow conditions (flood backwater and low-water navigation)

Name of study	Office authorizing study	Where conducted
Lock and dam No. 2, Miss. River, Hastings, Minn.[7]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Lock and dam No. 15, Miss. River, Rock Island, Ill.[6]	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Coefficients for Taintor and roller-gate dams[6]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Lock and dam No. 2, Kiskiminitas River, Pa.[6]	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Marmet lock and	Huntington Dist.	U. S. Engr. suboffice.

Name of study	Office authorizing study	Where conducted
dam, Kanawha River, Marmet, W. Va.[7]		Iowa City, Iowa
Roller-gate coefficients of typical arrangements of 3 gates	Rock Island Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Roller-gate coefficients of Miss. River dams 5, 5-*A*, and 8	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Winfield lock and dam, Kanawha River, Winfield, W. Va.	Huntington Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 5, Fountain City, Wis.	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 26, Alton, Ill.	St. Louis Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Ohio River, Montgomery Island lock and dam	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, Keokuk lock	Miss. River Power Co. (private)	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, lock and dam No. 4, Alma, Wis.	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Tenn. River, Pickwick lock and dam	Nashville Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Tenn. River, Chickamauga lock and dam	Nashville Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Cofferdam expt., Bonneville dam	2nd Portland Dist.	Linnton hyd. lab.
River model, Bonneville dam	2nd Portland Dist.	Linnton hyd. lab.
Three-mile rapids	2nd Portland Dist.	Linnton hyd. lab.
Monongahela River, new lock and dam No. 4[6]	Pittsburgh Dist.	Carnegie Tech.
Tuscaloosa dam	Mobile Dist.	Carnegie Tech.
Ohio River, Gallipolis lock and dam	Huntington Dist.	Carnegie Tech.
Coney Island Dike[6]	Cincinnati Dist.	U. S. Waterways Expt. Sta.
Ohio River dam No. 37[6]	Upper Miss. Valley Div.	U. S. Waterways Expt. Sta.
Chain of rocks[7]	St. Louis Dist.	U. S. Waterways Expt. Sta.

(*D*) Effect of dams on channel configurations (alignment of channel, and silting)

Name of study	Office authorizing study	Where conducted
Miss. River lock and dam No. 5, Fountain City, Wis.[7]	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River lock and dam No. 26, Alton, Ill.[6,7]	St. Louis Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Monongahela River, new lock and dam No. 4[7]	Pittsburgh Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, lock and dam No. 26, Alton, Ill.	St. Louis Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Miss. River, lock and dam No. 3, Red Wing, Minn.	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
White Water River, silting study	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Mill Creek diversion dam, 2nd model	2nd Portland Dist.	Linnton hyd. lab.
Scour below Beartraps[7]	Louisville Dist.	Louisville and Portland canal dry dock
Ohio River, dam No. 37[6,7]	Upper Miss. Valley Div.	U. S. Waterways Expt. Sta.
Starved Rock lock[6]	Chicago Dist.	U. S. Waterways Expt. Sta.
Ohio River, Coney Island dike[6,7]	Cincinnati Dist.	U. S. Waterways Expt. Sta.

3 Open-River Regulation for Navigation

(*A*) By submerged sills

Name of study	Office authorizing study	Where conducted
Backwater effects from submerged sills[6]	Chief of engrs.	Univ. of Michigan
St. Clair River[6]	Chief of engrs.	U. S. Waterways Expt. Sta.
Head of Passes[6,7]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
South Pass flume study[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
South Pass model[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
Fort Chartres, Miss. river[6,7]	St. Louis Dist.	U. S. Waterways Expt. Sta.

(*B*) By wing dikes (permeable and impermeable groins)

Name of study	Office authorizing study	Where conducted
Ohio River, Walkers bar[6]	Louisville Dist.	U. S. Waterways Expt. Sta.
Ohio River, Raleigh bar[6]	Louisville Dist.	U. S. Waterways Expt. Sta.
Miss. River, Stewarts bar[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Island No. 9[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Island No. 35[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Racetrack towhead, 1st model[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Hotchkiss bend[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Robinson Crusoe Island[6]	Memphis Dist.	U. S. Waterways Expt Sta.
Miss. River, Island No. 21[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Cat Island[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Island No. 20[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss River, Point Pleasant[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss River, Head of Passes[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
Miss River, Fort Chartres[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Fort Chartres second model[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Brooks Point[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Ste. Genevieve Bend[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Southwest Pass[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
Savannah River[6]	Savannah Dist.	U. S. Waterways Expt. Sta.
General dike study	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Head of Passes special tests[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
Miss. River, Chain of Rocks	St. Louis Dist.	U. S. Waterways Expt. Sta.
Ohio River, Pryors Island	Louisville Dist.	U. S. Waterways Expt. Sta.
Miss. River, Dogtooth bend	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Swift Sure towhead	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Grand Tower	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Wyanoke to Cow Island[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Southwest pass[6]	1st New Orleans Dist.	U. S. Waterways Expt. Sta.
Miss. River, Redman Point dike study[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Hopefield Point[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Wyanoke Island[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Morrison towhead[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
St. Johns River, Fla.[6,7]	Jacksonville Dist.	U. S. Waterways Expt. Sta.

(*C*) By longitudinal dikes

Name of study	Office authorizing study	Where conducted
Del. River, Deepwater	Philadelphia Dist.	Fort Mifflin lab.

Name of study	Office authorizing study	Where conducted
Point Range[6,7]		
Miss. River, Racetrack Towhead, 2nd model[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Fitler bend[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Chain of Rocks	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Swift Sure towhead[7]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Grand Tower[7]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Southwest Pass[6,7]	1st N. O. Dist.	U. S. Waterways Expt. Sta.
St. Johns River[6,7]	Jacksonville Dist.	U. S. Waterways Expt. Sta.

(*D*) By bank stabilization

Name of study	Office authorizing study	Where conducted
Miss. River, Cat Island[6,7]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Duckport to Delta Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Savannah River[6,7]	Savannah Dist.	U. S. Waterways Expt. Sta.
Miss. River, Delta Point eddy study[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Millikens bend[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(*E*) By dredging

Name of study	Office authorizing study	Where conducted
Ohio River, Walkers bar[6,7]	Louisville Dist.	U. S. Waterways Expt. Sta.
Miss. River, Island No. 35[6]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Racetrack towhead, 1st model[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Natchez Island to Glasscock Point[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Rifle Point[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Robinson Crusoe Island[6,7]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Point Pleasant[6,7]	Memphis Dist.	U. S. Waterways Expt. Sta.
Miss. River, Paw Paw bend to Yucatan cutoff[6,7]	Lower Miss. Valley	U. S. Waterways Expt. Sta.
Miss. River, Diamond Point, Yucatan Point[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Buckridge Crossing[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Racetrack towhead, Diamond Point[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River Memphis depot[7]	Memphis Dist.	U. S. Waterways Expt. Sta.
Ohio River, Pryors Island[7]	Louisville Dist.	U. S. Waterways Expt. Sta.
Miss. River Waterproof cutoff	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. river, Hopefield Point dredging[6]	Memphis Dist.	U. S. Waterways Expt. Sta.

4 Flood Control

(*A*) By levees and levee realignment

Name of study	Office authorizing study	Where conducted
Brunswick levee extension[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Natchez levee setback[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Helena to Donaldsonville[7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Kansas City	Missouri River Div.	U. S. Waterways Expt. Sta.
Atchafalaya Basin[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Ark - White separation[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(*B*) By channel straightening

Name of study	Office authorizing study	Where conducted
Miss. River, Greenville cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, cutoffs, Greenville to Red River Landing[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Slough Landing Neck cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, cutoffs at Leland, Worthington, Willow, and Marshall Points[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, cutoffs, Tarpley and Leland Necks[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, channel capacity study, Leland Neck to Pt. Breeze[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Dredged cutoffs[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Diamond Point cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Atch. River, Cow Island cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Natchez Island to Glasscock Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Waterproof cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Morville Landing to Esperance Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Rifle Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Atch. River, Lake Long bifurcation[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, American cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Cracraft towhead[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Paw Paw bend to Yucatan cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Diamond Point, Yucatan Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Buckridge Crossing[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Racetrack towhead, Diamond Point[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Togo Crossing[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Natchez Island[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Helena to Donaldsonville[7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Buckridge Crossing, 2nd model	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Kansas City[7]	Missouri River Div.	U. S. Waterways Expt. Sta.
Miss. River, Leland Neck cutoff[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Point Bar dredging[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, channel-capacity study vicinity Ark. City fuseplug levee[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(*C*) By diversions and outlets

Name of study	Office authorizing study	Where conducted
Birds Point, New Madrid floodway[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Atchafalaya enlargement[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Tests on experimental ditch	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Atchafalaya Basin[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Miss. River, Helena to Donaldsonville	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Ark - White separation[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

Name of study	Office authorizing study	Where conducted
Flood waves in diversion channels	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Boeuf and Atchafalaya pilot channels[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bonnet Carre spillway (3 models)	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

5 Coastal Harbors and Beaches

(*A*) Improvement of tidal estuaries and harbors for navigation

Columbia River estuary, 1st study[6]	2nd Portland Dist.	U. S. Tidal Model lab., Berkeley, Calif.
Columbia river estuary, 2nd study	2nd Portland Dist.	U. S. Tidal Model lab., Berkeley, Calif.
Beach-erosion investigation[7]	2nd Portland Dist.	U. S. Tidal Model lab., Berkeley, Calif.
Delaware River and Bay[6]	Philadelphia Dist.	Fort Mifflin
Deepwater Point Range[6]	Philadelphia Dist.	Fort Mifflin
Crosscurrents in channels[6]	Mobile Dist.	Spring Hill, Ala.
Dike location Columbia river	North Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
Relocation of ship channel, Columbia River	North Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
Effect of closing west channel into Bakers Bay	North Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
Effect of jetties at west channel, Bakers Bay	North Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
Galveston Bay	Galveston Dist.	U. S. Waterways Expt. Sta.
Chesapeake & Delaware Canal[6,7]	Philadelphia Dist.	U. S. Waterways Expt. Sta.
Maracaibo Bay	Standard Oil Co. of New Jersey	U. S. Waterways Expt. Sta.
Absecon inlet	Philadelphia Dist.	U. S. Waterways Expt. Sta.
St. Andrews Bay[6]	Gulf of Mexico Div.	U. S. Waterways Expt. Sta.
Brazos-Santiago Pass[6]	Galveston Dist.	U. S. Waterways Expt. Sta.
Winyah Bay[6]	Charleston Dist.	U. S. Waterways Expt. Sta.
St. Johns River[6]	Jacksonville Dist.	U. S. Waterways Expt. Sta.
Tidal river dredging	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Aransas Pass[6]	Galveston Dist.	U. S. Waterways Expt. Sta.
Mare Island	San Francisco Dist.	U. S. Waterways Expt. Sta.

(*B*) Breakwaters for reduction of wave action

Cellular-type steel-sheet pile breakwaters[6]	Milwaukee Dist.	Caisson plant, Milwaukee Harbor, Wis.
Wave action on various types of concrete caisson and rubble-mound breakwaters[6]	Milwaukee Dist.	Caisson plant, Milwaukee Harbor, Wis.
Air-breakwater expts.[6]	Milwaukee Dist.	Caisson plant, Milwaukee Harbor, Wis.
Port Washington harbor[6]	Great Lakes Div.	U. S. Waterways Expt. Sta.

(*C*) Inlets, sandy coasts, and beaches

Name of study	Office authorizing study	Where conducted
Effect of vertical seawall on sand movement[6]	Beach-Erosion Board	U. S. Beach - Erosion Board wave tank, Fort Belvoir, Va.
Sand movement using a natural beach sand[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Formation of a barrier, Beach No. 1[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Formation of a barrier, Beach No. 2[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Sand movement under wave action No. 1[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Sand movement under wave action No. 2[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Sand movement under wave action No. 3[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Sand movement under wave action No. 4[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Sand movement under tides and wave action[6]	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Beach-erosion investigation	2nd Portland Dist.	U. S. Tidal Model lab., Berkeley, Calif.
Ballona Creek	South Pacific Div.	U. S. Waterways Expt. Sta.

6 Miscellaneous Projects

(*A*) Improvement of model methods

Instruments[6]	St. Paul Dist.	U. S. Engr. sub-office, Iowa City, Iowa
Racetrack towhead bed-material study[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Effects of distortion[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Current-meter study	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Roller-type wave machine[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Slope distortion[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Mannings "n" in models[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bentzel velocity tube[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Control of algae in models[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Diaphragm orifices[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Erodible-bank study[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Design of weirs[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Salt-solution velocity method	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Directive energy grain sorting[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Directive energy sand feeding[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-load studies on synthetic sands[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Investigation of light-weight bed materials for use in models	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.

(*B*) Floating-plant designs

Dredge-pump model	North Pacific Div.	Linnton hyd. lab.
Towboat stability studies[6]	2nd New Orleans Dist.	River front, New Orleans
Experiments with pipe-line mixers	Memphis Dist.	U. S. Waterways Expt. Sta.

(*C*) Wave studies

Wave-pressure tests[6]	Beach-Erosion Board	Caisson plant, Milwaukee, Wis.
Study of wave force	Beach-Erosion Board	U. S. Beach-Erosion Board wave tank, Fort Belvoir, Va.
Wave action[6]	Lake Superior Dist.	Duluth, Minn.

(*D*) Stream-load studies

Transportation of bed material, Passamaquoddy tidal-power project[6]	Eastport Dist.	Alden hydraulic lab.
Transportation of bed load by Columbia River	North Pacific Div.	U. S. Tidal model lab., Berkeley, Calif.

Name of study	Office authorizing study	Where conducted
Photoelectric cell for measuring per cent of solids[6]	North Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
Local excesses of bed material[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bifurcating flume[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-load studies, natural sands[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-load diversion[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-load studies, synthetic sand mixtures[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Laconia bend[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Transylvania chute[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Critical tractive force of coarse material[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Design of sediment traps	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-material survey Miss. River, Cairo, Ill. to Gulf of Mexico and Atchafalaya River[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Sediment study of Atchafalaya Basin[6]	2nd New Orleans Dist.	U. S. Waterways Expt. Sta.
Bed-material survey at Montgomery cut-off, White River[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Bed-material survey of principal tributaries to Miss. River[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Sediment studies Miss. River prior to 1930[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Sediment studies Miss. River. 1930–1931[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Sediment studies Miss. River 1931–1932[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
(*E*) Protection of stream banks		
Revetment study	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Tetrahedral blocks[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Broken concrete and gravel revetment[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Articulated concrete mattress	Memphis Dist.	U. S. Waterways Expt. Sta.
(*F*) Earth-embankment studies		
Erosion tests at railroad embankment[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Investigation of levee and borrow-pit material, Big Lake reservation, Ark.[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Effects of overflow on railroad embankments[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Investigation and suggested design of material for a levee, Mauvais Terre levee and drainage dist.[6]	St. Louis Dist.	U. S. Waterways Expt. Sta.
Design of side slopes for embankment for site of marine hospital at Memphis, Tenn.[6]	U. S. Treasury Dept.	U. S. Waterways Expt. Sta.
Investigation of borrow-pit material for levees at Tulsa, Okla.[6,7]	Memphis Dist.	U. S. Waterways Expt. Sta.
Investigation of soil, Murray dam near Ardmore, Okla.[6]	WPA of Okla	U. S. Waterways Expt. Sta.
Investigation of pressure developed by plastic debris from mining operations[6]	South Pacific Div.	U. S. Waterways Expt. Sta.
(*G*) Flow in closed conduits—general		
Pipe bends[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Comparative tests of transportation of Newport Bay and Sacramento River sands in pipe lines[6]	South Pacific Div.	U. S. Tidal Model lab., Berkeley, Calif.
(*H*) Fluid mechanics		
Vortex tests	St. Paul Dist.	U. S. Engr. suboffice, Iowa City, Iowa
Velocity distribution and roughness coefficients[7]	Boston Dist.	M.I.T.
Helicoidal flow[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Slope distortion[6,7]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Investigation of limits of Red River backwater[6]	Lower Miss. Valley Div.	U. S. Waterways Expt. Sta.
Investigation of limits of the Illinois backwater[6]	Lower Miss. Valley	U. S. Waterways Expt. Sta.
(*I*) Canal studies		
East mooring basin, Cape Cod Canal[6]	Boston Dist.	M.I.T.
Velocity distribution and roughness coefficients[6]	Boston Dist.	M.I.T.
Determination of locus of high- and low-water profiles, Cape Cod Canal	Boston Dist.	M.I.T.
Velocity observations on progressive improvement of Cap Cod Canal	Boston Dist.	M.I.T.
Effect of dikes in the improved project, Cape Cod Canal	Boston Dist.	M.I.T.
Freeport Canal crossing, first phase[6]	Galveston Dist.	U. S. Waterways Expt. Sta.
Freeport Canal crossing, second phase[6]	Galveston Dist.	U. S. Waterways Expt. Sta.
Chesapeake and Delaware Canal[6]	Philadelphia Dist.	U. S. Waterways Expt. Sta.

American Hydraulic-Laboratory Practice

By LESLIE J. HOOPER,[1] WORCESTER, MASS.

It is the purpose of this paper to describe briefly representative American hydraulic laboratories and their work in recent years. Laboratories in this country and Canada were visited by the author during 1934 and 1935 as a Freeman scholar. It was obviously impossible for the author to visit every laboratory in this country in the time allotted and just as impossible to adequately describe each laboratory. Hence, considerable detail must be omitted in this article and many laboratories will not receive the attention their work deserves. Such omissions are not intentional but due only to the limitations of time available for inspection and space available for this paper.

SO FAR as is known to the writer, American laboratory work had its real start with James B. Francis who was working with the Proprietors of the Lowell Locks and Canals. The company owns the water rights of the entire Merrimac River at Lowell and sells water-power privileges to various manufacturing companies. The need of accurate water measurements caused Francis to make his experiments on contracted and suppressed weirs. This work was done in 1848, 1851, and 1852 on weirs 1.8 to 10 feet long (1).[2] The heads were measured with hook gages and the discharges determined volumetrically. This work was done with a high degree of precision as modern engineers who have duplicated his work can testify. This investigation was followed later by experiments on the flow of water in expanding tubes (1854).

Following Francis, his assistant and successor, Hiram F. Mills, made fundamental tests on piezometers and then on the flow of water in pipes (2). Still later, in 1888, John R. Freeman, an assistant of Mr. Mills, performed his calibration experiments with fire-hose nozzles using pitot tubes to traverse the jet (3). He demonstrated that the pitot tube was a reliable velocity-measuring instrument with a coefficient very close to unity.

Since the water-power privileges at Lowell were sold on an input basis, it was to the interest of the manufacturer to install the most efficient water wheel obtainable. This, of necessity, caused considerable interest in water-wheel design and it was at Lowell that the mixed-flow water wheel which is typical of modern design, was developed by Swain, Boyden, and Francis.

The next hydraulic laboratory of prominence to be installed was the Holyoke water-wheel testing flume at Holyoke, Mass. This was established in 1874 by James Emerson who had been trained at Lowell. The testing flume was taken over in 1882 by the Holyoke Water Power Company with Clemens Hershel in charge of the experimental work. At this laboratory, water wheels are tested under a head of from 12 to 18 ft in a flume 20 ft square in section and about 25 ft high. The water flows from the water wheel into the tailrace flume which is 35 ft long and 20 ft wide, with a sharp crested weir at the downstream end. The output is measured by a prony brake and direct-connected revolution counter. This test flume has been in operation continuously since its installation and it is still available for water-wheel testing. At present its facilities are not so frequently used since all water-wheel manufacturers have their own laboratories for routine testing.

The foregoing description offers very briefly a little of the historical background of hydraulic test work in this country.

Tables

In connection with the work of hydraulic laboratories at the present time, three tables have been prepared. Table 1 contains a list of manufacturers' and of Government laboratories which are engaged in a commercial form of research, together with an indication of the particular type of experimental work being done. Table 2 contains a list of college and university laboratories which are used for instruction, and scientific and commercial research. Table 3 lists those laboratories which are used for instruction and scientific research. In connection with Table 3, it must be borne in mind that college laboratories are used primarily for the instruction of undergraduates and graduate students since hydraulic experiments are practically always given with elementary hydraulic instruction, and postgraduate work often entails hydraulic investigations. At the time these visits were made a business depression had been much in evidence for five years. This has resulted in the heavier loading of professors' and instructors' time so that there has been little opportunity for research by college staffs. It is probable that had the inspection been made at a more favorable time, much more scientific research would have been found in progress.

TABLE 1 COMMERCIAL AND GOVERNMENT LABORATORIES

	Water wheels	Cavitation	Pumps	Calibration of instruments	Flow of water	Model hydraulic structures	River laboratories	Transportivity	Towing tanks
Bureau of Reclamation—Denver Office						x		x	
Fort Collins						x			
Montrose						x	x	x	
Byron Jackson Pump Company			x						
Department of Agriculture, F. C. Scobey					x				
Holyoke Water Power Company	x								
I. P. Morris Div. of Baldwin Southwark	x		x						
James Leffel Water Wheel Company	x								
Lowell Locks and Canals				x	x				
National Hydraulic Laboratory				x	x	x		x	
Newport News Shipbuilding and Dry Dock Co.	x	x	x						x
Pelton Water Wheel Company	x								
Pennsylvania Water and Power Co., Holtwood	x	x							
Rodney Hunt Machine Company	x								
Shawinigan Experimental Turbine Testing Plant	x	x							
S. Morgan Smith Company	x								
U. S. Army Engineers, Portland District Office	x					x	x		

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Hydraulic Division and presented at a meeting of The American Society of Mechanical Engineers, held at Niagara Falls, N. Y., September 17–20, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

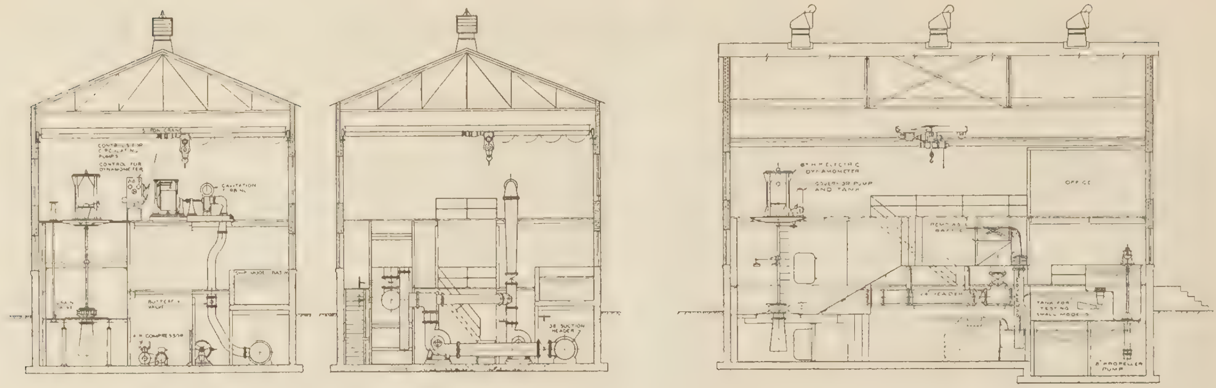


FIG. 1 SECTIONAL ELEVATION OF THE HYDRAULIC LABORATORY OF THE NEWPORT NEWS SHIPBUILDING AND DRY DOCK COMPANY AT NEWPORT NEWS, VA.

WATER-WHEEL LABORATORIES

Today, instead of all manufacturers bringing their experimental wheels to one place for test purposes, they all have their own laboratories except the Holyoke Machine Company which uses the facilities of the Alden hydraulic laboratory. All of the manufacturers' laboratories visited were busy at the time of the inspection. In general, the type of work being done was the improvement of design details, such as runner-blade shapes, casing details, draft tubes, and cavitation effects. Due to the commercial nature of their work, reports of their tests are rarely available.

As an example of a water-wheel manufacturer's laboratory, that of the Newport News Shipbuilding and Dry Dock Company at Newport News, Va., is briefly described.

TABLE 2 COLLEGE LABORATORIES ENGAGED IN RESEARCH AND COMMERCIAL WORK

	Water wheels	Cavitation	Pumps	Calibration of instruments	Flow of water	Model hydraulic structures	River laboratories	Transportivity	Towing tanks
California Institute of Technology			x	x		x	x		
California, University of			x	x		x	x	x	
Carnegie Institute of Technology						x	x		
Case School of Applied Science						x			
Cornell University				x	x				
Iowa, University of				x	x	x	x	x	
Maine, University of					x	x			
Massachusetts Institute of Technology		x				x	x	x	
Michigan, University of			x						x
Minnesota, University of						x		x	
Ohio State University				x					
Pennsylvania, University of				x		x			
Stevens Institute of Technology									x
Wisconsin, University of				x	x	x			
Worcester Polytechnic Institute	x		x	x	x	x	x	x	

TABLE 3 COLLEGE LABORATORIES USED FOR INSTRUCTION AND SCIENTIFIC RESEARCH

Columbia University
Illinois, University of
McGill University
Michigan State College
Oregon State College
Pennsylvania State College
Princeton University
Purdue University
Queen's University
Rensselaer Polytechnic Institute
Sheffield Scientific School of Yale University
Toronto, University of
Tufts College

NOTE: All college laboratories have equipment for the calibration of venturi and orifice meters, pitot tubes, and weirs. Practically all have pump and water-wheel testing equipment. Princeton, Purdue, and Rensselaer have towing flumes and carriages.

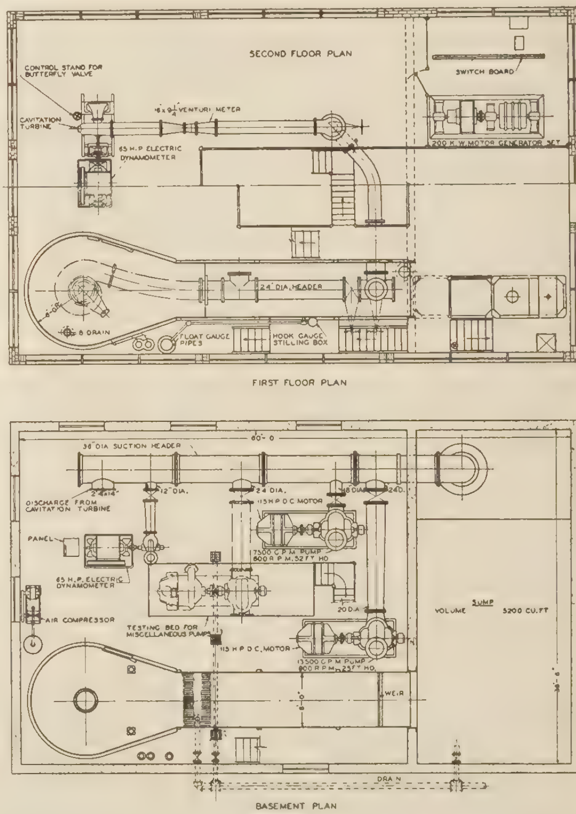


FIG. 2 PLAN VIEW OF THE HYDRAULIC LABORATORY OF THE NEWPORT NEWS SHIPBUILDING AND DRY DOCK COMPANY

The laboratory building, shown in Figs. 1 and 2, is $36^1/_2 \times$ 60 ft inside in plan and three stories in height, including the basement. Water is taken from a 5200-cu ft sump through a 36-in. suction header by two 115-hp electric motor-driven centrifugal pumps. One of these pumps delivers 22 cfs against a dynamic head of 25 ft for the water-wheel testing flume. The other supplies the cavitation unit with a flow of 16.7 cfs under a dynamic head of 52 ft. Both units may be used together to supply 40 cfs to the water-wheel flume.

In the basement there is also located a test bed for the ready installation and test of centrifugal pumps, having discharge

pipes as large as 24 in. in diameter. The power to drive these pumps is taken from a 65-hp horizontal electric dynamometer which, at the time of the inspection, was being used on the cavitation stand.

The water-wheel testing flume is an open circular steel tank, 12 ft in diameter. An opening in the bottom is provided with various rings so that any size model may be readily fitted in place, the standard size being about 16 in. in diameter. For open-flume settings, the water is pumped into the head-race tank 6 ft wide, 7 ft deep, and 20 ft long, connecting with the top of the circular tank. The discharge from the wheel passes from the 12-ft circular pit, under the tank, into a flume 6 ft wide, 8 ft deep and 22 ft long. Crushed-rock racks are installed and 6-ft suppressed weirs are located at the end of each of these flumes so that either pumps or water wheels may be readily tested without any appreciable change in the apparatus. To facilitate the testing of model water wheels in closed settings, a 24-in. pipe is installed between the discharge line of the pump and the side of the circular steel flume. The scroll case of the model is bolted to this pipe connection.

The total head on the water wheel is measured in one reading by means of two float gages, one connected to the tailrace flume bearing a scale, while the other, connected to the headrace flume or to the entrance to the scroll case, carries a pointer.

The head on the weir is measured by means of a float gage with a vernier allowing readings to be made to 0.001 ft.

The output of the water wheel is measured with a vertical 65-hp electric dynamometer which has a special double-ball thrust bearing. The intermediate race of this bearing is rotated in either direction by a small electric motor, allowing the effect of friction in the stator thrust bearing to be eliminated.

A master clock, through a program machine, automatically sends out signals for timing the test runs and through relays operates the revolution counter at the beginning and end of the run.

A 2500-ft-lb Woodward governor is installed in connection with the water-wheel testing flume and is primarily used in testing propeller-type turbine models with automatically adjustable vanes.

The cavitation equipment consists of a semiclosed circulating system. The water from the pump passes up to the second story through a diverging pipe to a converging elbow. In the horizontal run from this elbow, there is a $16 \times 9^1/_4$-in. venturi meter, from which the water passes to the inlet of the 10-in. water wheel. The draft tube from the water wheel passes through a rectangular pressure-regulating butterfly valve to the 36-in. suction header about 25 ft beneath in the basement.

The inlet pressure is regulated by means of the pump speed. Inlet and discharge pressures are measured with mercury manometers. Flow is measured with the venturi meter.

The output of the water wheel is measured in the same way as in the main water-wheel test flume except that a horizontal electric dynamometer is used. The most interesting detail of this unit is the use of a 10-in. glass cylindrical connection extending from just below the runner to the top of the draft tube which permits the observation of cavitation phenomena by means of a rotoscope or by stroboscopic light Other draft-tube models are provided with two or more observation windows.

Pump models, up to 10 in. eye diameter, are tested in the basement using the 65-hp horizontal dynamometer and the water is measured by 6- and 10-in. venturi meters. This equipment is cross-connected with one of the permanent circulating pumps so that special models may be tested both as pumps and as turbines.

In running each test, only two men are employed ordinarily; one man handles all adjustments and controls, which are conveniently grouped, while the other directs the test and computes and plots the results.

On the first floor a small steel flume 8 ft long, 4 ft wide and 4 ft deep is used for the testing of 4-in. draft-tube models. A small motor-driven propeller-type pump supplies 2.5 cfs under a head of 4 ft for this work. The discharge from the model tube passes into a steel flume 4 ft wide, 4 ft deep and 12 ft long, with a 2-ft contracted weir at the far end. The model draft tube is tested with straight and swirling flow as a head regainer, pressure measurements being taken of the head water, tail water, and at the throat of the tube.

There is also a small towing tank on the first floor of the laboratory, 8 ft wide, 4 ft deep and 56 ft long. It is designed to accommodate a 4-ft ship model. The ship model is towed by a cord which, in turn, is operated by a gravity dynamometer. The greatest care was taken in the design of the dynamometer to secure the same bearing and friction losses at all times. The speed of the model is obtained by means of a photoelectric cell, operating on the spokes of the idler pulley of the system. All data are automatically recorded on a chronograph.

Other laboratory equipment consists of a 200-kw motor-generator set, an air compressor, a 20-in. engine lathe, a 16-in. Alden dynamometer, and a 5-ton crane which is arranged to serve each of the three floors. A separate storage building is provided in the rear of the laboratory.

Cavitation Testing

Due to low-head developments, the trend in recent years in water-wheel design has been to larger units and higher specific speed so that cavitation has become a problem of increasing importance. This problem has been attacked in two ways, the first being to improve the contours of water-wheel blades and the second to find materials more resistant to the effects of cavitation.

There are several laboratories engaged in cavitation research from the viewpoint of water-wheel design, namely, the Shawinigan Experimental Turbine Testing Plant of the Shawinigan Water Power Company, the Holtwood Hydraulic Laboratory of the Pennsylvania Water and Power Company, and the laboratory of the Newport News Shipbuilding and Dry Dock Company which has been described. The Holtwood Laboratory located at Holtwood, Pennsylvania, is described here since the work done there is representative of this branch of water-wheel testing.

Water is taken from the forebay of the Holtwood plant, shown in Fig. 3, by a motor-driven centrifugal pump which can supply a discharge of 56 cfs at a head of 35 ft. The discharge from the pump flows into a head box which is provided with suitable racks and a sluice gate. This gate is operated by an electric motor and is remotely controlled from the laboratory, allowing the head in this forebay tank to be varied over a range of about 20 ft. A bellmouthed intake, 52 in. in diameter, connects with a 36-in. penstock, which delivers the water through a 36×15-in. venturi meter to the rectangular pressure box at the laboratory. The wheel pit is 12×18 ft in plan and 9 ft high, allowing ample room for the installation of model scroll cases. Directly below the wheel there is an observation chamber in which elbow, spreading and conical draft tubes of the shape desired may be installed. The draft tube under test at the time of the visit was fitted with small windows which allowed the inspection of the wheel and of the draft-tube flow while the unit was in operation. This has been done in the past using steady and stroboscopic light. The draft tubes discharge into a closed box which has a regulating gate at the bottom and a vacuum pump taking off at the high point. This makes it possible to keep the draft tubes sealed and yet, by proper manipulation of the gate, the pressure in the box may be dropped to a very low equivalent water surface. By varying the pressure in the forebay tank and in the draft-tube

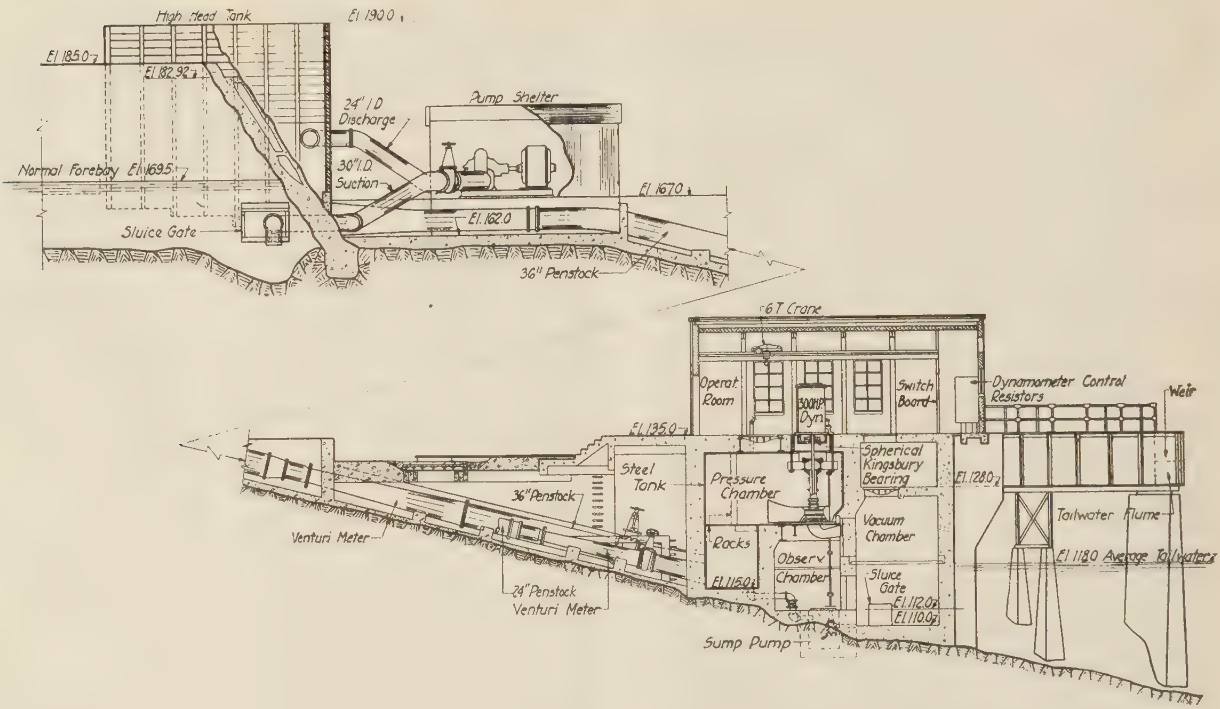


Fig. 3 The Holtwood Hydraulic Laboratory of the Pennsylvania Water and Power Company

chamber tests can be run with a range of heads from 66 ft to 30 ft. For cavitation tests σ can be varied from 0.4 to 1.5.

The pressure head on the wheel is measured with a barometric type of mercury manometer. The tailrace elevation is read by means of a counterweighted float gage. These two gages are set close together allowing one observer to take both readings which, when subtracted, give the net pressure head on the wheel. The discharge is measured by the 36 × 15-in. venturi meter which was calibrated by the Allen salt-velocity method.

The output of the water wheel is measured by a vertical 300-hp electric dynamometer, which is equipped with the same form of special thrust bearing which was described in connection with the Newport News Shipbuilding and Dry Dock Company laboratory. The speed of the wheel is counted by a Leeds and Northrup synchronous timer driven by a small a-c generator which, in turn, is gear-driven from the dynamometer shaft. The clutch of the timer is operated automatically by contacts on an International time clock which also controls the warning and reading bells.

The Holtwood laboratory was used initially to determine the characteristics of water wheels for the Safe Harbor power plant, both from the design viewpoint and finally from the operation viewpoint when the power output as limited by cavitation was determined for various head conditions (4, 5). The laboratory has also been used by various manufacturers for their experimental work in cavitation.

There are a number of laboratories in the country which have equipment for the determination of the resistance of various materials to the effects of cavitation. Such cavitation stands are in operation at Holtwood, Safe Harbor, University of Toronto, and at the Massachusetts Institute of Technology.

The cavitation test stand at the Massachusetts Institute of Technology, shown in Fig. 4, consists of a closed recirculating system with a 4-in. square venturi section in which the pressure, temperature, and velocity of the test fluid can be controlled. Water is supplied by two electric motor-driven centrifugal pumps having a capacity of 13.3 cfs at 60-ft head when operating in parallel, and 6.7 cfs at 120-ft head when operated in series. The discharge from the pumps enters the end of a vertical cylindrical steel tank about 4.5 ft in diameter and 10 ft high which is equipped with baffles and racks to produce quiet flow conditions. The

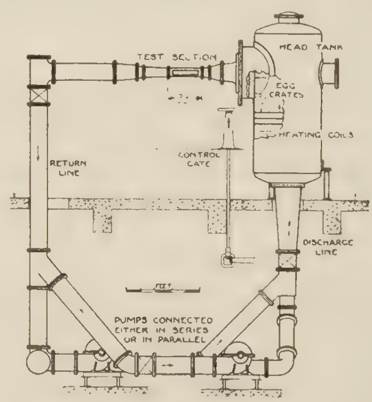


Fig. 4 Arrangement of Apparatus for Cavitation Research at Massachusetts Institute of Technology

side outlet from this tank reduces to the venturi throat section in which cavitation occurs. The discharge from this section passes to the suction side of the pumps. Cooling coils are installed in the pressure tank so that the temperature of the fluid may be regulated. Connections are provided so that the pressure in the tank and at the test section may be readily determined and provision is made to determine the air content of the fluid during the test. Plate-glass side walls are provided for the test section to facilitate observation and the photographing of the cavitation phenomena. High-speed motion pictures have been taken of

the apparatus in operation. The results of these tests have been published in progress reports from time to time (6).

A new form of cavitation apparatus has been designed recently at M.I.T. to determine the resistance of materials. It consists essentially, of a metal rod which is set into longitudinal vibration by means of an electrical oscillator circuit. The frequency of the vibration is in the neighborhood of 7000 cycles per sec. The specimen to be tested is mounted on the end of the vertical rod and immersed about $^1/_2$ in. in a bath of the fluid which is to be tested with the specimen. The fluids which have been used to date are alcohol, salt water, and fresh water. The metal rod and the bath are maintained at the correct temperatures by means of cooling coils. The effect on the specimen seems to be identical with that obtained in the venturi-shaped testing sections. However, a series of tests is now in progress to correlate the results of these two methods of test procedure.

Pump Testing

As in the case of water wheels, practically all of the pump manufacturers have their own laboratories which are used both in their design work and in the routine testing of pumps. In addition, many of the colleges are equipped to test pumps using an input up to 10 hp. The most satisfactory method of measuring the power is by means of an electric dynamometer, although, in some cases, torsion dynamometers have been used with success. In a few cases a calibrated electric motor has been used. The pressures at the inlet and the discharge of the pump are usually measured with mercury manometers for moderate pressures and with calibrated Bourdon-tube gages for high pressures. The discharge is usually determined either by weighing, volumetric tanks, or by calibrated venturi meters. Possibly, the outstanding pump-testing laboratory built in recent years is that of the California Institute of Technology at Pasadena, Calif., the description of which follows.

The pump-testing laboratory at the California Institute of Technology, shown in Fig. 5, was built with the cooperation of the metropolitan water district of Southern California for the purpose of determining the best type characteristics and then the best commercial design of pump to be used in bringing water from the Colorado River to Los Angeles.

The equipment is set up in a room about 20 ft wide, 50 ft long and three stories high, with a basement beneath. The suction for the pump is taken from a tank 4 ft in diameter and 25 ft high. The water is discharged through a bank of four venturi meters of different sizes and thence, through a throttling valve, back to the suction tank. Inasmuch as this is a closed system, the suction pressure may be adjusted to any desired value by means of auxiliary pumps, allowing cavitation tests to be made very readily. The range in suction pressure is from about —25 ft of water to a positive pressure of about 125 ft of water. The four venturi meters are controlled by means of grease-sealed plug valves so that only one meter is used at any given time and that during the stable range of the meter. These venturi meters have been carefully calibrated by calibrated volumetric tanks. The throttling plug valve is motor-operated and serves to regulate the discharge pressure of the pump. This valve is remotely controlled from the operator's desk.

In order to keep tests of different pumps on a truly comparative basis, the pressure piezometers at the inlet and discharge of the pump are located at the same position relative to the pump and tank and in the same pipes for all pumps tested.

The suction and discharge pressures are measured by means of specially designed and constructed gages operating on the principle of the dead-weight gage tester. These gages have a range from a vacuum to 500 lb per sq in. and are sensitive to 0.01 lb per sq in. Two of these gages are used to measure the suction pressure and the discharge pressure. The same type mechanism is adapted to weigh the mercury differential of the venturi meters.

The input to the pumps is determined by means of a horizontal electric dynamometer which has a capacity of about 400 hp at 5000 rpm.

The torque of the dynamometer is measured by a combination of a hydraulically loaded piston and a weight which slides transversely on the dynamometer itself. This apparatus is sensitive to the nearest 0.25 in-lb of torque.

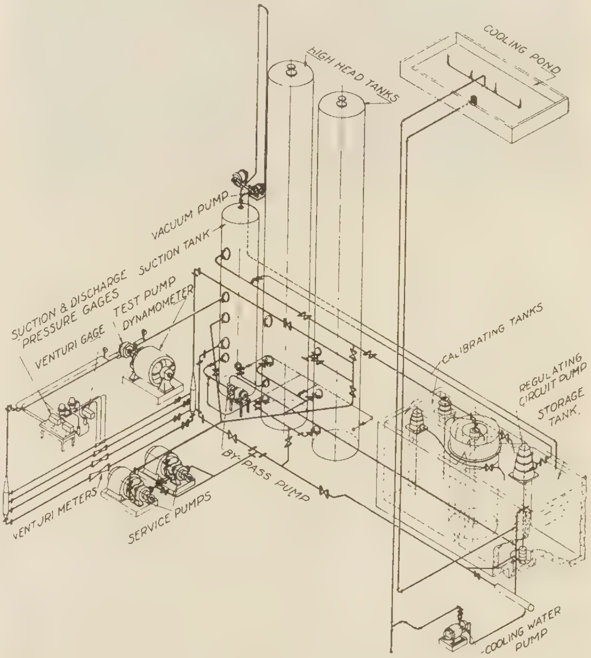


Fig. 5 Pump-Testing Laboratory at the California Institute of Technology

The speed of the dynamometer is held constant by an automatic speed regulator controlled from a standard time source. A special gearbox allows the speed of the pump to be adjusted in 0.5-rpm steps up to 5000 rpm and the system is so constructed that the speed is kept at any chosen value to one part in one hundred thousand.

It is thus seen that the speed and pressure conditions for a test are selected and the dynamometer weight, venturi deflection, discharge and suction pressures are read from the automatic balancing scales and gages. Since all the gages and apparatus are balanced automatically, there is no personal error introduced into any of the readings. The sensitivity and the accuracy of the apparatus are such that whole series of tests can be made with all test points within 0.1 per cent of the true values.

In addition to the suction pressure tank, there are two steel tanks 5 ft in diameter and 40 ft high. These tanks are used to test the pumps under abnormal operating conditions determining the performance of the unit under conditions of power failure and sudden changes in pressure conditions. A long test program was in progress for the metropolitan water district of Southern California at the time of the inspection. Some idea of the type of test work being done may be obtained from the brief papers mentioned in the Bibliography (8, 9, 10). A detailed description of this test equipment will probably be published shortly.

Calibration of Instruments and Apparatus

Practically all college laboratories are equipped to calibrate hydraulic measuring instruments to some extent. A few examples of the facilities available and the work being done are indicated in the following brief description.

There are a number of tangent-current-meter rating stations which are used to calibrate current meters and pitot tubes, among which are the tanks at the Colorado Agricultural Experiment Station, Rensselaer Polytechnic Institute, Princeton University, and that of the Bureau of Standards. The last is probably the best known of the group and the description follows.

Fig. 6 Circular Current-Meter Rating Station at the Alden Hydraulic Laboratory

The still-water channel in which the current meters are rated is 400 ft long, 6 ft wide, and 6 ft deep. The towing car travels on steel rails mounted on the sides of the channel. The cast-iron wheels of the car and the surface of the track are ground to a smooth finish. When the tracks are clean the operation of the car is vibrationless. The car is equipped with a 5-hp electric motor which, in turn, drives a Waterbury hydraulic gear which is connected to the wheels of the car. Thus the car may be driven at any desired speed up to 12 fps in either direction. All of the controls and the recording apparatus are located on the car.

The current-meter contacts operate a relay which pulls over a notched wheel by means of a magnet. Mounted on this notched wheel are several contacts so located that one operates at the start of a test and the others operate after 50, 100, or 150 current-meter impulses.

The distance traveled by the car during the test is indicated by a pair of darts, which are discharged by springs at the beginning and end of the run. These darts are shot into a wooden scale which is mounted on the floor beside the track. The shooting of the darts is controlled by the contactor in the current-meter relay circuit.

The time of the test is measured by a Leeds and Northrup synchronous timer which is operated on a special constant-frequency line from the radio laboratory nearby. The clutch of the timer is operated by the contactor in the current-meter relay circuit.

This laboratory is used most frequently in calibrating current meters for commercial purposes.

At the Alden Hydraulic Laboratory of the Worcester Polytechnic Institute there is installed a circular rating station which has been used for the calibration of current meters and pitot tubes. This station, shown in Fig. 6, consists essentially of a horizontal steel boom 84 ft long, pivoted at its center and rotated by a water wheel through a rope drive. The pond in which the meter is towed is from 6 ft to 10 ft deep, of irregular outline and is large enough so that any disturbed water passes out from the measuring circle between passages of the instrument. The speed of rotation is maintained constant by means of a centrifugal governor equipped with a friction brake. The maximum speed obtainable at the end of the boom is about 25 fps.

Venturi meters and pitot tubes are calibrated at a great number of college laboratories and at the National hydraulic laboratory. At the University of Pennsylvania work may be done in any size pipe up to 16 in. The water which is taken from a stand pipe 6 ft in diameter and 40 ft high is discharged into two weighing tanks of 16,000-lb capacity each. These tanks are used in succession so that continuous measurements of the discharge are possible allowing a high degree of precision to be obtained.

At the Alden Hydraulic Laboratory, shown in Figs. 7 and 8,

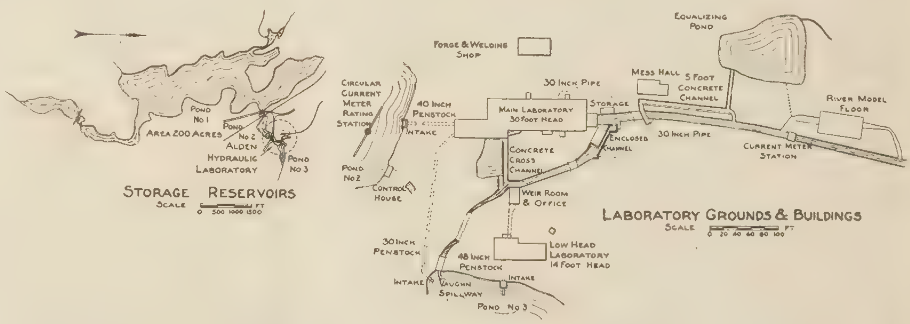


Fig. 7 Laboratory Grounds and Buildings of the Alden Hydraulic Laboratory at Worcester Polytechnic Institute

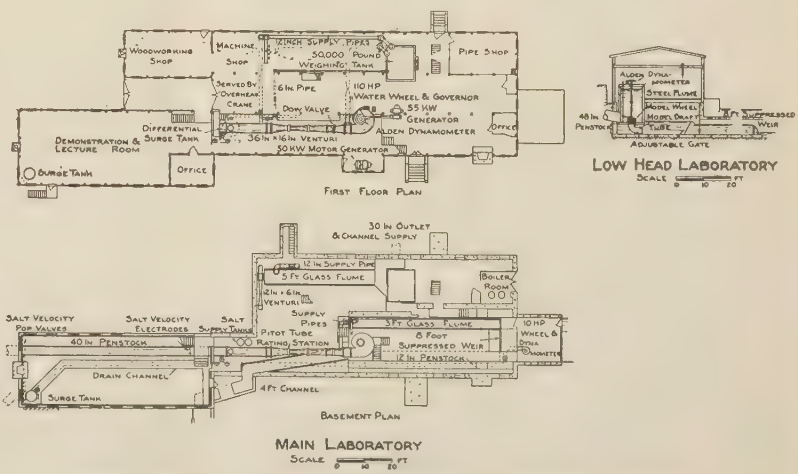
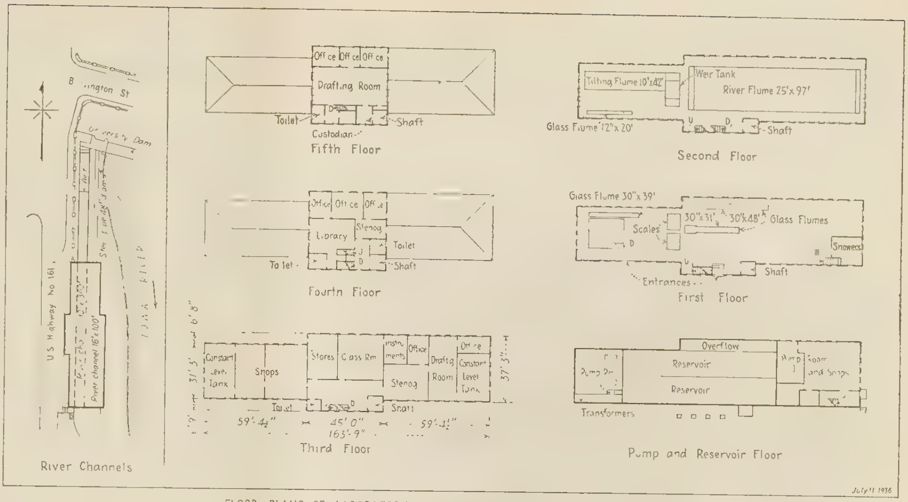


Fig. 8 Alden Hydraulic Laboratory at Worcester Polytechnic Institute



FLOOR PLANS OF LABORATORY - IOWA INSTITUTE OF HYDRAULIC RESEARCH
State University of Iowa, Iowa City

Fig. 9 Plan of Hydraulic Laboratory at Iowa State University

venturi meters and weirs are calibrated by means of a single weighing tank of 50,000-lb capacity. The results of a long series of weighing-tank calibrations of a weir were published recently (14).

Large venturi meters have been checked in place in the field by N. R. Gibson using his pressure-time method and by Prof. C. M. Allen using the salt-velocity method.

At the National hydraulic laboratory, venturi meters may be conveniently calibrated by means of large volumetric tanks having capacities up to 35,000 cu ft.

In general, weirs are not considered measuring instruments of the highest precision in this country unless calibrated as used, because the effect of the velocity of approach may so easily change the calibration. Possibly the best known research work in this branch of hydraulics is the report of the long series of weir calibrations made at Cornell University by Schoder and Turner (12). The discharges were measured volumetrically in a standpipe 6 ft in diameter and 60 ft high.

At the Massachusetts Institute of Technology a series of investigations concerning the flow and pressure conditions in the nappe of a weir were made by Dr. Hunter Rouse and analyzed from the fluid-mechanics viewpoint (15).

There has been considerable work done in recent years upon the calibration of sharp-edged orifices installed in a pipe line. At the Ohio State University a comprehensive investigation was made in cooperation with two engineering societies and a commercial company upon the flow of water through orifices (16). The discharge was checked in large volumetric tanks having capacities of about 7500 cu ft. The differential pressure was measured with mercury manometers for high heads, with water manometers for moderate heads and with differential hook-gage manometers for the very low heads. The effect of the location of the orifice in the pipe, the thickness of the orifice plate, and the effect of the location of the pressure taps were determined.

This investigation was supplemented by an investigation at the Case School of Applied Science where a series of tests was made at very low Reynolds numbers using various heavy oils (17).

There has been installed at the laboratory of the Worcester Polytechnic Institute a traveling screen for the measurement of the mean velocity in a concrete channel 5 ft wide, 5 ft deep, and 120 ft long. It is possible to check the discharge by means of the large weighing tank.

Flow of Water in Pipes and Channels

There are many laboratories in this country which are interested in the flow of water in pipe lines and open channels. The interest in this subject is evidenced by the large number of papers and articles published recently on this branch of hydraulics. At the hydraulic laboratory of the University of Iowa, shown in Fig. 9, Yarnell and Nagler have reported upon the flow of water in 6-in. pyralin elbows (18). These elbows were made in various shapes, one being a standard short 6-in. pipe elbow. The flow was studied with threads, dye, and air bubbles and the coefficient of loss and the velocity and pressure distribution were determined at the same time. A similar investigation was made at the Oregon State College by Mockmore. Prof. Ernest Schoder at Cornell University has recently completed an investigation of the losses in pipe bends of different radii, tees, and crosses.

Prof. Bakhmeteff at Columbia University has done considerable work on the flow of water in open flumes (19, 20). His mathematical deductions were checked by experiments in a glass-sided tilting flume 20 ft long, 22 in. deep, and 6 in. wide.

In connection with the irrigation work of the Department of Agriculture in the western part of the country, F. C. Scobey has done considerable work determining the losses in open channels and flumes over a wide range in velocity (22, 23, 24, 25, 26). All of this work was done on the large irrigation canals after installation, the discharge being checked by current-meter measurements and by the color-velocity method.

At the Alden Hydraulic Laboratory, Professor Allen has made a study of the dispersion of salt solution in flowing water in a penstock (21). The salt solution was introduced under a low pressure through a trailing pitot tube and the spread of the salt solution was determined by means of a grid of small electrodes located at a cross section in the pipe. The salt solution was introduced at various distances upstream from the grid.

At the University of Illinois, an investigation is in progress upon the flow of water in glass pipes 2 in. in diameter (27). The velocity distribution in the pipe is being determined by taking motion pictures of oil particles in suspension in the water which are illuminated at right angles to the camera with a flat beam of light. The preliminary published results of this investigation indicated an accuracy of the velocity measurements of the order of 2 per cent.

A study of the flow in vertical sewer pipes is being made at the University of Wisconsin and at the National Hydraulic Laboratory (30). Since the pipe is not full of water, the flow phenomena are considerably different from the more ordinary cases.

At the Universities of Purdue and Wisconsin, a series of investigations are now in progress on the flow of fluids through sharp-edged orifices and over sharp-crested weirs. It is the purpose of these investigations to express the flow in terms of the fundamental properties of the fluids, i.e., viscosity, density, and surface tension. The hydraulic laboratory at Purdue University is shown in Fig. 10.

Model Hydraulic Structures

Probably the most popular branch of hydraulic-laboratory experimental work at the present time is the testing of model hydraulic structures, nearly all college laboratories being equipped to perform these tests. In general terms, all that is required for this type of work is a glass-sided flume, a suitable water supply, and a means of measuring the water used. The most common model ratios range from 1:20 to 1:60. Practically all hydraulic structures installed at the present time have had the details of design checked by model tests but reports are rarely published due to the commercial nature of the work. Some of the work which has been done recently may be indicated by the title.

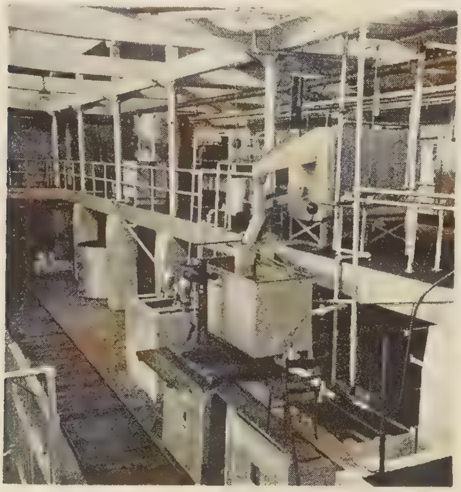


Fig. 10 Hydraulic Laboratory at Purdue University

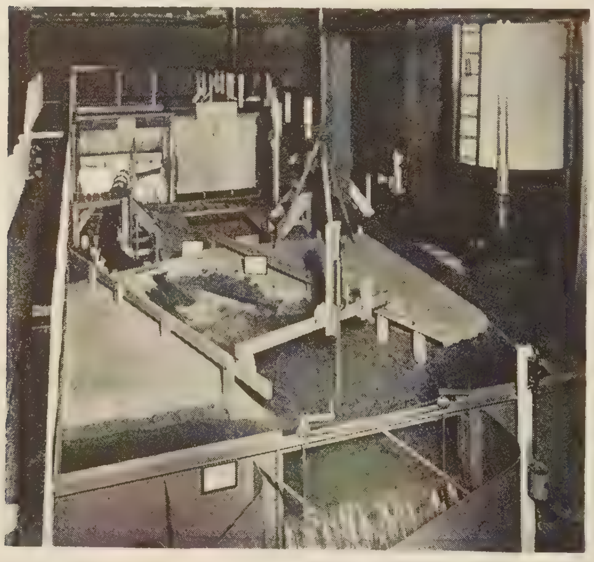


Fig. 11 Testing Flume at the Case School of Applied Science

At the Denver Office of the Bureau of Reclamation, tests were in progress on the Taylor Park Side Spillway, the Caballo Dam, the Imperial Valley Diversion Dam and the Caballo Tunnel Outlet. At the Fort Collins laboratory of the same organization, a number of tests were in progress to determine erosion conditions, the pressure distribution on the face of the dam and the operation of a drum crest gate for the Grand Coulee dam on the Columbia River.

At the University of Iowa a number of studies were in progress on the characteristics of dams, gates, locks, and stilling pools to be used in connection with the canalization of the Upper Mississippi River.

Professor Straub of the University of Minnesota had two tests in progress on dams of the Northern State Power Company, studying erosion and pressure distribution.

Professor Barnes at the Case School of Applied Science working in conjunction with the U. S. Army engineers of the Zanesville district has just completed a series of tests of eleven different dams and stilling pools for the Muskingum project. One of these is shown in Fig. 11.

At the Alden Hydraulic Laboratory, tests have been made of the rock-fill dam, the navigation lock and the filling gates of the Passamaquoddy tidal-power project, erosion and pressure studies of the Holtwood Dam and hydraulic details of the Barkhamstead reservoir for the city of Hartford.

While not as recent as the models previously mentioned, a very interesting model was built at the Carnegie Institute of Technology in connection with the Chute à Caron Dam on the Saguenay River (31). As a result of the tests a precast concrete dam was successfully tipped into place in the river bed effecting the final closure of the diversion dam.

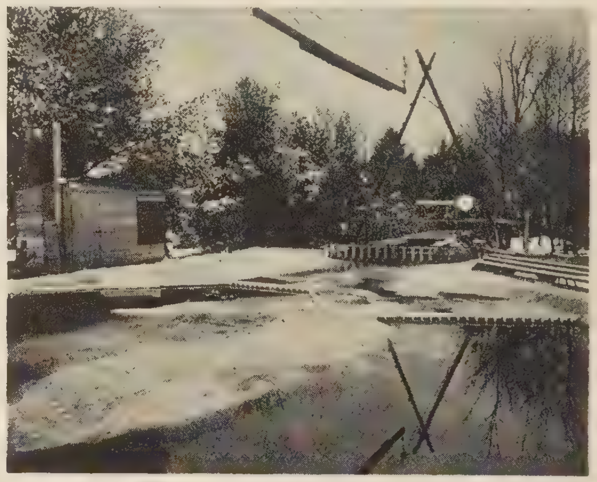


Fig. 12 Rock Island Plant Model at the Alden Hydraulic Laboratory

River Laboratories

Models of sections of rivers and canals have been quite widely used in recent years. There are two general types of river models: The first, in which a uniform scale is used, and the second, where a distorted scale is used. The first type of model is adapted only for a detail study of a section of the river where the scale ratio may be kept small, maintaining suitable velocities and depths in the model. Where it is necessary to study longer reaches of the river or where the river is shallow the use of the distorted model is necessary. Only a few examples will be given illustrating each type of these models.

At the Alden Laboratory a 1:100 undistorted model was made of the Columbia River at the site of the Rock Island Dam. The model proper, shown in Fig. 12, was 34 ft wide and 64 ft long so that a reach of the river of about 1.2 miles was represented. The purpose of this model was to determine the performance of the spillways, necessary tailrace excavation, height of cofferdams,

necessary cofferdam removal, the head on the plant during construction and after completion, the handling of ice and many other problems in connection with the design and operation of the plant.

At the Linnton Laboratory of the Army engineers of the Portland district there has been built an undistorted 1:100 model of the Columbia River, for the reach including the Bonneville dam. This model, shown in Fig. 13, is 325 ft long and of varying width as determined by the topography of the river. This model is being used to determine about the same details that were stated for the Rock Island model test. In addition, that portion of the model river bed downstream from the spillway may be made movable instead of fixed so that the formation of bars and the erosion of the riverbanks may be studied.

At the Montrose laboratory of the Bureau of Reclamation a 1:40 uniform scale model of the Colorado River at the Imperial Valley diversion dam was being constructed at the time of the inspection. The Colorado River at this point is relatively wide and shallow and carries a heavy load of material in suspension and along its bed. It was the purpose of this model to determine

Fig. 13 Columbia River Model at the Linnton Laboratory

the proper design of inlet and spillway works to avoid serious silting effects.

The second class of models of rivers using a distorted scale is much more common. The outstanding example of the use of distorted models is undoubtedly the United States Waterways Experiment Station at Vicksburg, Miss. This laboratory and its work have been ably described by Lieutenant Falkner. A large number of distorted models have been used at the laboratory of the University of Iowa in connection with the canalization of the Mississippi River. At the time of the inspection a model having a horizontal scale of 1:500 and a vertical scale of 1:100 was being tested in a river flume 25 ft wide and 97 ft long. This model shown in Fig. 14, had a fixed bed and the purpose was to determine the velocity distribution and the hydraulic gradient of the river in the vicinity of the lock and dam.

A model of the San Gabriel River mouth at the Alamitos Bay inlet was found in operation at the California Institute of Technology (32). The vertical scale was 1:60 and the horizontal scale 1:120. The purpose of this model was to study methods of protecting the beach and to conserve an adequate river channel to the sea. The mechanism for producing the tides was very simple. The inflow to the model was set at a value equal to the maximum rate of flow of the tide. The discharge valve was automatically operated by a time clock and had sufficient capacity to discharge all the water admitted by the inlet valve and, in addition, drop the water surface at the required maximum rate of ebb of the tide. In order to have proper conditions for the study of the beach problem, a wave-making machine was installed at the seaward side of the model and was designed to provide waves of the desired frequency and magnitude.

At the University of California a distorted model of the mouth of the Columbia River was being tested to determine a stable and navigable channel for the river at and near its outlet to the

Fig. 14 River Model and Flume at the University of Iowa

Fig. 15 Columbia River Model at the University of California

sea (46). This model, shown in Fig. 15, was 72 ft long and 42 ft wide and built to a horizontal scale ratio of 1:3600 and a vertical scale ratio of 1:64. The tides were controlled by a motor-driven cylindrical valve which provides a model duplication of the tides in nature. The effect of the upper reaches of the river and its tributaries which were affected by the tidal movement was duplicated by means of a labyrinth which was calibrated so as to provide the proper velocities and hydraulic gradients at the upstream end of the model.

At the Massachusetts Institute of Technology there is being tested at the present time a model of the Cape Cod Canal. The horizontal scale is 1:600 and the vertical scale is 1:60. This model, shown in Fig. 16, is 115 ft long and 34 ft wide. The model was first constructed in December, 1934, to represent the present

canal. The hydraulic gradients existing in the model under various conditions were compared to the field data obtained on the actual canal. Having thus checked the performance of the model, the width of the canal was increased to correspond to the designed bottom width of 500 ft in the field. The effect of the proposed changes was determined together with possible improvements in the design to assist navigation.

An interesting feature of this test was the use of a new form of gage for determining the water elevation in the model canal without interfering with the flow. A vacuum-tube circuit was used, the change in condenser effect in the grid circuit causing changes in the current in the plate circuit which were read with a milliammeter. The condenser was formed by a plate fixed over the surface of the water and the water surface itself, with the air spacing between acting as the dielectric. The use of these gages permits practically simultaneous readings of the water level at a number of different points in the model, which is advantageous in determining the hydraulic gradient. The movable weirs at each end of the model which controlled the tidal effects, were operated to maintain the correct water levels by means of these gages working with a cam which was laid out from the observed

Fig. 16 Model of Cape Cod Canal at Massachusetts Institute of Technology

tidal cycle in nature. A more detailed description of this model will be published shortly.

An unusual distorted river model has been constructed at the Carnegie Institute of Technology under the direction of Prof. H. A. Thomas for the study of flood waves in rivers. The vertical scale is 1:80, the width scale is 1:4000, and the longitudinal scale is 1:20,000. As built, the model has the proper areas and volumes but the proper hydraulic roughness is attained by allowing transverse fins of sheet metal to project into the flowing water. The hydraulic gradient of the model is adjusted under steady flow conditions to agree with that of the prototype by trimming the metal fins. One section of the model which corresponds to a reach of 60 miles of river has been tested with satisfactory results.

Transportivity

With movable-bed models, the question of transportivity of material is of the first importance. A number of laboratories are working on this problem both in determining the fundamental laws of the phenomena and the action of bed movement in specific cases. The latter case usually is found in the river-model work which was described previously. The more fundamental work is usually done with some specific sand in tilting flumes, determin-

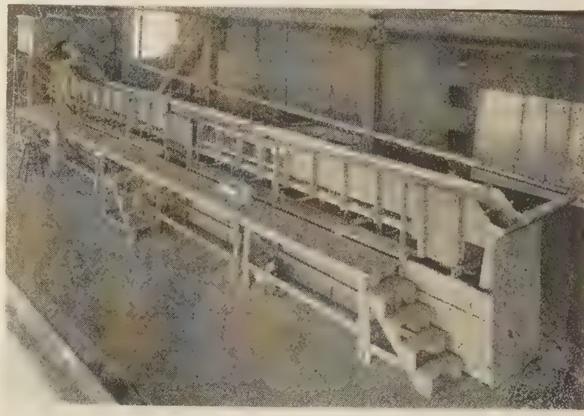


Fig. 17 Glass-Side Tilting Flume at the National Hydraulic Laboratory

ing the rate of movement of material under uniform flow conditions and adding material at the upstream end of the flume at the same rate that it is being removed at the lower end of the flume so as to maintain the amount of the material in the flume bed. Work of this nature was found in progress at the National hydraulic laboratory, the Montrose laboratory of the Bureau of Reclamation, the University of Iowa, University of Minnesota, Worcester Polytechnic Institute, and at the Massachusetts Institute of Technology. The type of work being done may be gathered by recent publications on this subject. The work at the Alden laboratory was of a somewhat different nature since, in this case, the study concerned the building of a dam in tidal flow with rocks dumped from barges and cableways. The problem here was to determine the path of the material in water moving at various velocities and the proper shape of the structure to economically withstand the pressures developed. The 45-ft glass-side tilting flume at the National hydraulic laboratory is shown in Fig. 17.

A number of laboratories are engaged in research on percolation through soils, earth dams, and similar structures. These investigations are usually coupled with studies of the materials determining density, void ratios, cohesiveness, permeability, and such subjects which come under the general heading of soil mechanics.

Towing Tanks

The outstanding towing tanks for the testing of ship models are those operated by the government. At the United States Model Experimental Basin there is a towing tank designed for the use of 20-ft ship models. The tank is 20 ft deep, 44 ft wide and approximately 350 ft long. This equipment was installed about 1900 and for many years it was the largest in the world.

Recently a large towing tank has been installed by the United States government at Langley field. This tank is 24 ft wide, 12 ft deep and 2000 ft long. The maximum speed which had been attained at the time of the visit was about 58 mph, but that is not the maximum speed obtainable. Due to the use of rubber-

tired wheels which were finish-ground in place, this high speed was obtained with no perceptible vibration.

At the Stevens Institute of Technology and at the Newport News Shipbuilding and Dry Dock Company there are located tanks designed to accommodate 4-ft ship models.

A large towing tank adapted for 10-ft ship models was installed at the University of Michigan in 1906. This tank, shown in Fig. 18 is 300 ft long, 10 ft deep and 22 ft wide. A car, carried on steel rails, spans the tank and can tow a model at a maximum

Fig. 18 Towing Tank at the University of Michigan

speed of about 15 fps. The drag of the model is recorded directly on a moving-strip chart together with the distance the car has traveled and the elapsed time in seconds.

The models at this laboratory are cast in hard wax approximately to shape and then cut to size at 0.5-in. intervals of elevation in a special machine. Two sets of motor-driven cutters are mounted on a pantograph so that both sides of the model are cut at once. The cutter position is determined by the location of a tracing point which is moved by the operator over the full model size longitudinal horizontal sections.

The work done in this laboratory has been reported in the Transactions of the Society of Naval Architects.

Conclusions

In review, the following trends in laboratory practice seem to be apparent.

There is a distinct tendency toward the use of more elaborate apparatus, particularly automatic balancing and recording instruments. In the water wheel testing laboratories, for instance, a time clock is used to operate the revolution counter and provide, in addition, all the signals for a test run. In many cases the dynamometers are self-balancing. In the testing of ship models in towing tanks, all the data are recorded automatically. The use of automatic devices has been prompted by the desire for more accurate results and personal errors are thus largely eliminated.

In the next place, there seems to be a very definite trend toward the use of larger models or smaller scale ratios. Here again, the reason for this is undoubtedly the desire for increased accuracy of the test results. A number of instances of the use of large models may be cited, such as the 1:15 model of the Grand Coulee dam, shown in Fig. 19, and the 1:40 model of the Colorado River at the Montrose laboratory, the 1:100 models of the Columbia River at the Linnton laboratory and at the Alden hydraulic laboratory, and the 1:600 × 1:60 scale distorted model of the Cape Cod Canal at the Massachusetts Institute of Technology.

Finally, the laboratories in this country have increased much more rapidly in numbers than in size in recent years. That is to say, the development of laboratory work has not given us a few very large laboratories but rather a larger number of smaller installations. In some instances, small laboratories have been installed for the model tests of a single project. This development is usually prompted by the desire to maintain very close contact between the engineering design department of the project and the laboratory where the model work is being done.

It is believed that this close contact is necessary for the best results. However, this same close contact can be effected by having a responsible engineer of the project working on the model tests at a laboratory best equipped for the particular type of test work regardless of the geographical location. This arrangement allows the full realization of the value of the experience of the laboratory director, both in laboratory technique and in

Fig. 19 Model of the Grand Coulee Dam at the Montrose Laboratory of the Bureau of Reclamation

field hydraulic problems. It is believed that this phase of the problem is too frequently forgotten.

In conclusion, it is seen that the college and commercial laboratories of this country and Canada are well equipped for investigations in all phases of hydraulic engineering. The research work found in progress included all branches of the science in spite of the fact that the financial situation had not been at all favorable for this type of work for several years previous to the inspection.

The inspection trip upon which this paper is based was made possible by a John R. Freeman Scholarship granted by the Boston Society of Civil Engineers and by funds which were provided by the Alden Hydraulic Laboratory of the Worcester Polytechnic Institute.

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History and Present Status of Research and Specifications of Diesel Fuel Oil

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The authors refer to the need for specifications and a classification of Diesel fuel oils, and discuss what has been accomplished in this respect by the various organizations interested in the subject. The following properties of fuel oils are discussed to clarify the meaning of each in the specifications of fuel oils: Ignition quality, viscosity and pour point, carbon residue and ash, water and sediment, sulphur, and flash point. A review of objectives of fuel-oil research concludes the paper.

APPRECIATION of the need for specifications covering Diesel fuel oil first arose about 1928 and publication of efforts in this direction was made shortly thereafter (1).[3] Since that time much has been learned and, while the subject has been by no means exhausted, it is now fairly clear that the following known properties of fuel oil must be given attention not only in any attempt to set specifications but also in developing fuels and engines: (1) Ignition quality, (2) viscosity and pour point, (3) carbon residue and ash, (4) water and sediment (5) sulphur, and (6) flash point.

While investigation of any one of these properties will be concerned more or less with the others, it may be in the interests of clarity to discuss the various items separately.

Ignition Quality by Engine Test Methods

By the term "ignition quality" is meant the degree of ease with which the fuel ignites and the rapidity with which it burns when injected into the cylinder of a Diesel engine. By this very definition, this quality is frequently the most important of any of those listed. The Diesel cycle depends for ignition on the injection of fuel into a heated mass of air so that the significance of ignition quality is obvious. The realization of the importance of this property is not new. As far back as 1919 attempts were made to link this with spontaneous-ignition temperature as determined in a bomb (2). This work and subsequent bomb experiments have shown that the temperature required for ignition varies markedly with such factors as delay period, air-fuel ratio, atomization and turbulence.

Subsequently it has been found that the temperature required for ignition in actual engines also varies with such factors (3). For those who may not be acquainted with these terms, brief definitions are as follows:

1 Delay period is crank angle in degrees between the beginning of fuel injection and the point at which the pressure begins to rise rapidly as a result of combustion.

2 Air-fuel ratio is the ratio of the weight of air in the cylinder to the weight of fuel injected.

3 Atomization refers to the extent to which the fuel has been subdivided and dispersed in the process of injection into the cylinder.

4 Turbulence is the degree of agitation of the cylinder contents (first air and later mixture), during injection and combustion.

The modern laboratory engine method of investigating ignition quality dates from the experimental work of LeMesurier and Stansfield (3), Boerlage and Broeze (4), and Pope and Murdock (5). Boerlage and Broeze used a slow-speed engine and obtained variation in compression ratio by throttling the air inlet. Each fuel was compared with blends of cetene and mesitylene to determine the blend having the same delay angle. Later alpha-methylnaphthalene was substituted for mesitylene. Each rating was expressed as the percentage of cetene in the blend which matched the fuel under test. Pope and Murdock ran their tests using a modified CFR (Cooperative Fuel Research Committee) variable-compression knock-testing engine. The rating was expressed as "critical compression ratio," this being the lowest compression ratio at which the fuel being tested would just ignite under controlled conditions of speed, temperature, injection advance and pressure, fuel quantity, and inlet air.

Cooperation between these investigators further refined the engine testing method by improvements in the design of the CFR engine and the development of more exact apparatus for the determination of delay angle. A factor in the extension of this work was the formation of the so-called "Volunteer Committee for Compression Ignition Fuel Research," which is composed of members of the petroleum and Diesel-engine industries, actively engaged in experimental work with the modified CFR engine. The work of this committee is not yet finished, but great progress has been made (6).

At this point it is well to point out that different batches of cetene used by the volunteer group varied somewhat in ignition quality, either because they were composed of varying percentages of the several varieties of cetene or because of impurities.

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[1] Standard Oil Development Company, New York, N. Y. Dr. Becker was graduated from Marietta College in 1909 with an A.B. degree. In 1917 he received a Ph.D. degree in physics from Harvard University. He served as instructor in physics at Tufts College, 1911–1912 and as assistant in physics at Harvard University from 1914 to 1917. During the war he was associated first with the National Advisory Committee for Aeronautics and then with the oil and lubrication branch of the Air Service, serving as a private in the Air Corps during last six months. He was connected with the Aluminum Castings Company in 1919–1920 as physicist, and in 1920 entered the employ of the Standard Oil Company (N. J.) where his first work was in lubrication research. At present he is engaged almost entirely in fuel work both Diesel and gasoline.

[2] Secretary, Diesel Engine Manufacturers Association. Mem. A.S.M.E. Mr. Reed was graduated in 1917 with a B.S. degree in engineering from the University of Illinois. He served as first lieutenant in the 135th Aero squadron of the U. S. Army until 1919 at which time he entered the employ of the Ingersoll-Rand Company as sales engineer, later becoming engineer in the compressor engineering department, and assistant to the manager of the oil and gas-engine department. From 1926 to 1929 he was engineer in charge of construction for J. B. Eurell Company, Philadelphia, Pa. Since 1929 he has been with the Diesel Engine Manufacturers Association, as secretary.

[3] Numbers in parentheses refer to Bibliography at the end of the paper.

Contributed by the Oil and Gas Power Division and presented at the Ninth Annual Oil and Gas Power Meeting of The American Society of Mechanical Engineers, held at Ann Arbor, Mich., June 24–27, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1936, for publication at a later date. Discussion received after the closing date will be returned.

In its efforts to make pure cetene, The E. I. duPont de Nemours Company found it much easier to prepare pure *cetane*, a saturated hydrocarbon. Not only can this be more readily duplicated, but the committee members found that the ignition quality differed only slightly from that of cetene. Accordingly they have standardized upon *cetane* numbers rather than cetene numbers as had been originally proposed by Boerlage and Broeze.

Some investigators have been approaching the problem of engine ratings from a somewhat different angle, namely, that of determining the octane number of a blend of 25 per cent of the Diesel fuel under test and 75 per cent of a reference gasoline of known octane rating. The blending octane number of the Diesel fuel is then calculated from the rating of the mixture and that of the reference gasoline. It is claimed that such blending octane numbers correlate well with cetane numbers as determined by the delay method. One must keep in mind, however, that such ratings are in an inverse relationship to cetane numbers; that is, the lower the blending octane number, the better the ignition quality of the fuel oil (7).

A careful determination of cetane numbers by means of either the critical compression ratio or delay methods, involves a routine which requires considerable time. Apparatus has been developed at the Pennsylvania State College which gives promise of reducing the amount of time required per determination (8). There is also a possibility that the suggested method will result in greater accuracy. The apparatus aims to eliminate bouncing-pin difficulties of the delay method by the use of electrical apparatus for determining the required compression ratio of the test fuel for ignition at top dead center and a selected constant delay angle.

Ignition Quality by Fuel-Property Methods

The high cost of laboratory test engines and the specialized technical personnel required for their operation has led many investigators to attempt to find an index of ignition quality obtainable from fuel properties which are easily determined by routine laboratory methods. These investigators have developed several such indexes based upon two or more properties, such as viscosity, gravity, aniline point and distillation temperatures. One of the first of these indexes to combine two properties is the so-called Diesel index (9), which is perhaps the most widely used today. This index, as well as viscosity-gravity constant (10), the boiling-point gravity number (11), and U.O.P. characterization factor (7) are means for expressing the paraffinicity of fuels, since engine and field work have shown that, in general, the more paraffinic a fuel, the better its ignition quality. In Diesel index, aniline point as a means of expressing paraffinicity is corrected by gravity. Likewise gravity is used as a modifying factor for viscosity or boiling point, as the case may be, in the other indexes mentioned. For details see the original papers cited in the Bibliography. Such indexes cannot be expected to be indicative of the ignition quality of fuels to which chemical substances classified as "dopes" have been added. As a matter of fact, for doped fuels engine methods give values which are not in line with correlation curves for standard commercial fuels.

These indexes are already being used to a considerable extent, and the probabilities are that one of them will be adopted by common usage as a field method for estimating ignition quality even after an engine method has been standardized. However, the latter will always have to be the final reference.

Viscosity and Pour Point

Viscosity has an important bearing upon fuel nozzle efficiency and handling from the storage tank to the engine cylinder. Pour point must be considered along with viscosity in problems of storage and handling.

Viscosity, gravity, and surface tension are the oil properties which affect the hydraulics of injection and spray dispersion. Much experimental work has been conducted on injection systems and fuel sprays. In this country the work has been done principally at Pennsylvania State College and the Langley Field station of the National Advisory Committee for Aeronautics. The application of most of the findings has been on injection-system and engine design but it has been established that dispersion becomes more even as viscosity is decreased (12). More exact information could be used on permissible variations in viscosity without change, or with minor changes only, of injection-system adjustments.

Early engine experiments on the effect of viscosity were mainly concerned with finding the maximum value which could be utilized without impairing efficiency. With the advent of the high-speed Diesel engine, the emphasis is now shifted to a consideration of the allowable minimum viscosity. To meet higher requirements of ignition quality and cleanliness, fuel producers have furnished oils of lower viscosities. Some engine manufacturers contend that low-viscosity fuels cause nozzle and pump wear and reduction of pump volumetric efficiency. This contention is based almost entirely upon field results obtained under conditions where the effects of viscosity, if any, cannot be separated from the influence of dirt contamination. Experiments now being conducted may throw considerable light upon this subject.

Carbon Residue and Ash

Early investigation into the effect of fuel properties on engine operation laid great emphasis upon carbon residue as determined by the Conradson method. Inspired by the A.S.M.E. Progress Report (13), a number of manufacturers ran tests on fuels containing 3 to 6 per cent Conradson carbon in the years 1930 and 1931. Most of these tests were made using engines of relatively large cylinder dimensions, operating at what is today considered slow speed; the smallest engine used had a bore of 7 in. and was run at 650 rpm. The last tests along this line were conducted by the U. S. Navy at the submarine base at Groton, Conn., on an engine of about 10-in. bore running at 375 rpm (14).

Unfortunately, at the time when these experiments were conducted there was no practical means for determining the ignition qualities of the fuels tested or even much appreciation of the importance of this quality. Undoubtedly the test fuels did vary in ignition quality and therefore the test results were not conclusive because poor operation and increasing fuel consumption, attributed at that time to higher Conradson carbon, may have been due to poor ignition quality.

Diesel-engine manufacturers and oil refiners are agreed that limitation of Conradson carbon is still important. The exact limits for different sizes and types of engines are somewhat a matter of opinion although this, in turn, is based upon field experience. Some manufacturers and refiners feel that the quality of fuel which is supposed to be measured is not characterized accurately enough by the Conradson determination. Various other tests have been suggested such as the combustion residue test recommended by the Atlantic Refining Company (15). If the Conradson test for carbon is really not indicative, some other test ought to be developed and standardized. Then there should be some systematic experimental work devoted to finding practical limits for carbon content of fuels intended for various types and sizes of Diesel engines.

Carbon deposits in engines can be the result of causes other than Conradson carbon content. Some of these are poor spray

distribution, erratic injection and overloading, any one or more of which can give trouble even though the fuel may have low Conradson carbon.

The Conradson carbon determination serves as a check on the amount of high boiling-point fractions in the fuel such as distill above 700 F. The limit on the amount of fractions boiling above this point is still being actively investigated.

Ash in fuel oil is commonly believed to be a cause of cylinder wear. Refiners seem to experience no difficulty in producing practical fuels very low in ash content, so that there is today not much discussion along this line. It is commonly believed that dirt contamination is responsible for considerably more wear than can be attributed to ash content of the fuel as it leaves the refinery.

Water and Sediment

The ignition quality requirements of high-speed engines are such that refiners have been forced to produce fuels so high in general quality that only traces of water and sediment are present. The problem is to prevent contamination from the time fuel leaves the refinery until it is consumed in the engine. Just as the gasoline user has had to learn to keep water out of gasoline, so must the Diesel operator learn to protect his fuel against dirt contamination.

In so far as fuels for heavy-duty engines are concerned, there are few or no experimental data to show what effect, good or bad, water has when present in the usual amounts occurring in heavier fuels. The consensus is that it has no effect. It is generally known that sediment causes wear of fuel pump and other injection parts. Slow-speed heavy-duty engines, in general using higher-viscosity fuels, are less sensitive to this factor. Experiments now under way on the use of low-viscosity fuels should increase our knowledge in a quantitative way of the amount of dirt contamination that can be tolerated.

Sulphur

The effect of sulphur is a controversial subject. It has been known for some time that sulphur causes corrosion of steel exhaust lines and mufflers in intermittent and idling operation. On the other hand, numerous operators have used high-sulphur fuels in engines on more or less continuous duty without having observed any ill effects. This field experience comprises practically all the information on the subject.

Difficulties experienced in the oil fields where Diesel engines are operated on crude oils have been attributed by some to the sulphur content. The field data are such, however, that it is impossible to segregate any effect of sulphur from that of the fine sand which is almost always present in crude oil. In any case, forms of sulphur, mostly hydrogen sulphide, which have been blamed for these troubles do not occur in refined fuel oils.

Those who have made a study of Diesel fuels seem to be in agreement that sulphur does not present an important problem. There are still many users who are inclined to attribute all difficulties with fuel to its sulphur content. This point of view is a heritage from the past and will undoubtedly change when users become better acquainted with the experience of those who successfully use fuels of high sulphur content.

Flash Point

Apparently flash point is of little significance in so far as the operation of a Diesel engine is concerned. This is amply demonstrated by the fact that many engines in the oil fields successfully burn crude oils of low and high flash points. Flash point is of importance, however, because of fire hazards in storage and handling. Insurance regulations and state laws recognize this and include provisions for minimum flash point.

Recent U. S. Navy Experiments

While it is impossible to cover all of the valuable work in progress or recently reported on, no résumé would be complete without a mention of the extensive experimental work during the past two years conducted by the U. S. Naval Engineering Experiment Station, Annapolis, Md. This work is of special interest to the Diesel industry because a most important angle in so far as the Navy is concerned is to be able to decide upon one specification for fuel to be used in all its equipment. Conclusions to date are based upon data obtained from not only laboratory test engines but also commercial engines similar to those in actual use on naval vessels. A series of 25 fuels, varying widely in ignition quality, and from 33 to 57 sec Saybolt-Universal viscosity at 100 F, from minus 40 to plus 30 F in pour point, from 0.004 to 2.12 per cent carbon residue, from 0.06 to 1.03 per cent sulphur, and from 168 to 232 F flash point were investigated. The work is being extended to cover another series of twenty-five fuels. In the meantime no major conclusions have been drawn. Recommendations for Navy fuel specifications are given in Table 1.

TABLE 1 U. S. NAVY SPECIFICATIONS FOR DIESEL-ENGINE FUEL OIL

Flash point, closed-cup, minimum, F	150
Pour point, maximum, F	0
Viscosity, minimum, Saybolt Universal at 100 F, sec	35
Viscosity, maximum, Saybolt Universal at 100 F, sec	45
Water and sediment, maximum, per cent	0.05
Total sulphur, maximum, per cent	1.00
Carbon residue, maximum, per cent	0.20
Ash, maximum, per cent	0.01
90 Per cent distillation temp, maximum, F	675
Corrosion at 212 F	Negative
Diesel index number, minimum	45

American Society for Testing Materials Classifications

The demand for some classification of Diesel fuels was such that Technical Committee C of the American Society for Testing Materials appointed the Coordinating Subcommittee on Diesel Fuels early in 1934 with a commission to collect and classify data and to prepare tentative specifications if possible, and if not, some classification.

The membership of this subcommittee is composed of representatives of the A.S.T.M. Technical Committee C, the Volunteer Committee for Compression Ignition Fuel Research, the A.S.M.E., the S.A.E., Diesel-engine manufacturers and fuel producers. The subcommittee drafted its first classification which was published by the A.S.T.M. in 1934. This was revised in minor respects in 1935 (16) and the subcommittee recommended slight further changes at the June, 1936, meeting of Technical Committee C.

While present knowledge of the subject is too incomplete to permit of even tentative specifications, this classification has proved useful throughout the industry in an informative way. It is so widely known that it is not necessary to repeat it here. It is interesting to note that the recommended specifications of the U. S. Navy, previously mentioned, closely parallel the A.S.T.M. classification for a No. 1-D fuel.

Things to Be Done

From the point of view of engine design and performance, a satisfactory solution of the ignition-quality problem is essential. Such a solution will involve (*a*) the establishment of a standard laboratory-test method that will yield reproducible results and will correlate with performance in service, (*b*) the production of fuels of adequate ignition quality at a reasonable cost, and (*c*) design improvements that will make engines operate satisfactorily on a wider range of fuels. Satisfactory progress is being made by the Volunteer Committee, the U. S. Navy and others on (*a*), both individually and collectively. There is great

need for more correlation data between laboratory tests and commercial-engine performance, although some fuel producers have already made notable contributions. In so far as (*b*) and (*c*) are concerned, only the barest mention of possibilities is in order here. The increased adoption of supercharging, the resort to higher compression pressures, and refinements in injection-apparatus and combustion-chamber design will all, no doubt, contribute materially to the solution of the problem of ignition quality. At the present time the fuel-producers' problem is complicated in many districts by the impossibility of using the same fuels for Diesel engines and household burners and thus obtain reasonable distribution costs for the former. When high-speed Diesels are widely enough distributed to justify separate fuel-distributing facilities universally, special refining treatments, now available, can and will be used.

Of importance in connection with ignition quality is the subject of so-called "dopes," or chemicals to be added to fuel to improve its rating. Not only must a successful dope be effective in this respect, but it should be safe to handle and ship, easy to blend, and low in cost. The last requirement would seem to be essential in order to compete with improvement of fuel by refining methods or with wider acceptance of fuels by way of engine design.

Present discussion about viscosity is centered around what minimum value can be tolerated without detrimental effect to injection parts and on what viscosity range can be handled without major adjustment of fuel-injection systems. An accepted conclusion that low-viscosity fuels are entirely satisfactory would have the effect of materially widening the sources from which Diesel fuel can be drawn as well as minimizing the problem of low pour point.

There is a need for study of better methods for preventing contamination of fuels with dirt during storage and in handling. There should be more thorough analyses of field difficulties commonly attributed to dirt or low viscosity, as the case may be, to determine which is really responsible. Those who contend that sulphur limits should be reduced should obtain any present data substantiating their position. If the Conradson test is not the proper one to evaluate the effect of residual carbon, a more indicative test should be proposed and, if generally accepted, limits for carbon content by such a test should be established by laboratory and field experimentation. As these various problems are solved, specifications for Diesel fuel will logically develop from the A.S.T.M. classification.

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Square-Edged Inlet and Discharge Orifices for Measuring Air Volumes in the Testing of Fans and Blowers

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This is the third paper presented by the author to the A.S.M.E. on the flow of air in fan ducts. In the first paper,[2] it was established that the normal flow in such ducts is vortical in character and that the use of straighteners is necessary in order to obtain a reliable air-volume measurement. The second paper[3] contained proposals for (1) the adoption of a specified design of pitot tube as a standard in fan testing and (2) the use of square-edged inlet or discharge orifices for measuring air volumes except at or near maximum fan capacities. In the present paper, the results of an extended series of investigations on a simplified form of square-edged orifice are presented. The discharge coefficients obtained in these investigations are identical with those found in extensive German tests and adopted in the standard rules of the V.D.I. and by the International Standards Association. This agreement justifies the use of the coefficients for the large-sized ducts and the varieties of velocity distribution found in fan discharge ducts. The substitution of inlet and discharge orifice measurements for pitot-tube traverses and the adoption of these coefficients in a fan-testing code would, the author believes, greatly shorten and simplify the process of testing a fan and would make it more accurate.

IN A previous paper,[3] the author discussed the devices available for measuring the volume of air handled by a fan or blower. Comparing these various devices, the author arrived at the following conclusions:

1 Rounded nozzles are the most accurate but their cost and inconvenience make them impracticable.

2 Pitot-tube traverses can, in general, be made to give satisfactory measurements but demand much time for observations and may lead to appreciable errors when used with fans which have a strongly pulsating discharge.

3 Square-edged inlet and discharge orifices are as good as nozzles when their discharge coefficients are accurately known, and escape the disadvantages of pitot tubes. For a given size of duct, they can be used for a greater discharge capacity than nozzles and consequently permit the testing of fans for higher capacities than are possible when nozzles are used.

4 Duct orifices were dismissed as impracticable on account of the greatly increased length of duct demanded for their use.

At the time of writing last year's paper,[3] there was available in the literature one set of values of discharge coefficients for inlet and discharge orifices[4] which appeared to be accurate and applicable to the testing of fans. A more recent development is the third edition of the standard rules of the Verein deutscher Ingenieure for the measurement of flow of fluids through nozzles and orifices.[5] The two previous editions of these rules covered only the use of duct or pipe orifices, that is, orifices preceded and followed by a considerable length of straight pipe or duct. Inlet and discharge orifices were not included. The third edition[5] gives discharge coefficients and other data on inlet and discharge orifices. The work of Stach[4] is quoted as supplying data for such orifices, but the coefficients for discharge orifices are not those found by Stach but are the slightly lower values found by Witte[6,7,8,9] for duct orifices. These are the coefficients given in the V.D.I. rules and adopted by the International Standards Association. A tolerance of ±1.5 per cent is proposed for the recommended coefficients,[5] which is somewhat higher than the tolerance proposed by Stach.[4]

A comparison of these coefficients is shown in Fig. 1, which gives both the V.D.I. coefficients and the Stach coefficients; the broken lines indicate the tolerance limit above and below the V.D.I. values. It will be seen that it is only for area ratios m (orifice area/duct area) greater than 0.6 that the difference between the Stach and V.D.I. values exceeds 1.5 per cent. No justification is given in the V.D.I. rules for the selection of the Witte values in preference to the Stach values, but the membership of the committee which made the selection is such as to give

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[2] "Air Flow in Fan-Discharge Ducts," by Lionel S. Marks, Trans. A.S.M.E., vol. 56, 1934, paper PTC-56-2, pp. 871–877.

[3] "The Measurement of Air Flow in Fan Inlet and Discharge Ducts," by Lionel S. Marks, Trans. A.S.M.E., vol. 57, 1935, paper PTC-57-1, pp. 429–432.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

[4] "Die Beiwerte von Normdüsen und Normblenden im Einlauf und Auslauf," by E. Stach, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 78, 1934, pp. 187–189.

[5] "Regeln für die Durchflussmessung mit Genormten Düsen und Blenden," Deutsche Industrie Normen (D.I.N.) 1952 (German engineering standards), Verein deutscher Ingenieure, Berlin, third edition, 1935.

[6] "Durchflussbeiwerte der IG-Messmündungen für Wasser, Öl Dampf und Gas," by R. Witte, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 72, July-December, 1928, p. 1493.

[7] "Durchflusszahlen von Düsen und Staurändern," by R. Witte, *Technische Mechanik und Thermodynamik* (*Forschung auf dem Gebiete des Ingenieurwesens*), vol. 1 A, 1930, pp. 34, 72, and 113.

[8] "Die Strömung durch Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 2 A, 1931, pp. 245, 291.

[9] "Neuere Mengenstrommessungen zur Normung von Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 5 A, 1934, p. 205.

great weight to their decision. In addition, it may be stated that the experimental work of the author reinforces their decision.

The orifice shape, and the location and details of the pressure-measuring device, are carefully prescribed in the V.D.I. rules and are shown in Figs. 2 and 3. Fig. 2 shows acceptable arrangements of ring piezometers with openings at the orifice plate (*corner taps*). Fig. 3 is for larger sizes of duct and shows corner taps of drilled holes connected externally to an encircling pipe.

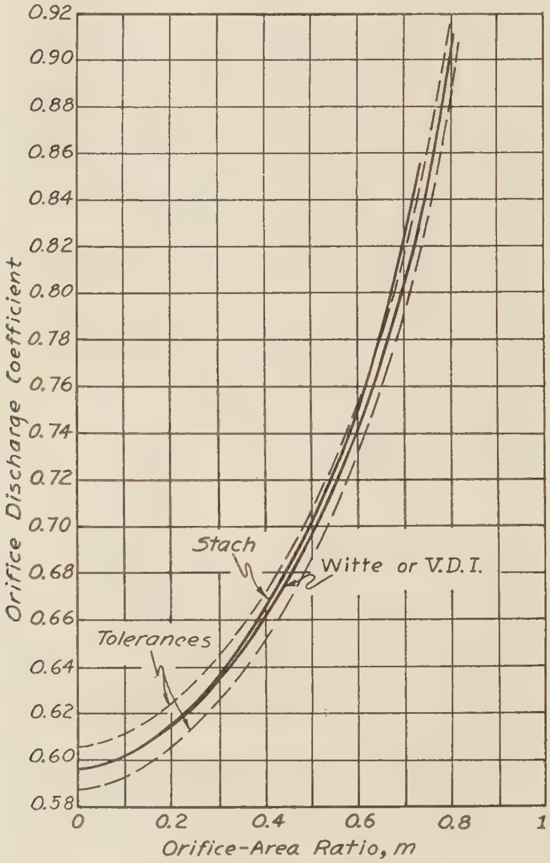

Fig. 1 Discharge Coefficients for Square-Edged Orifices Recommended in V.D.I. Rules Compared With Those Determined by E. Stach

For the sizes of duct common in the testing of fans, the ring-piezometer arrangement is complicated and costly and apparently offers no advantage over the use of the drilled hole connections such as are shown in Fig. 3. The use of four drilled holes, spaced 90 deg apart, has been investigated by the author. With air flowing in large ducts, with adequate length of duct, and with the use of straighteners to give a good velocity distribution across the duct, the differences between the static pressures measured at the four holes were found to be so small in all cases as to be negligible. When connected to a ring of small pipe, as shown in Fig. 3, the difference between the average pressure (so-called) and the pressure at any one hole was usually too small to be measured.

Discharge Orifices

The author's investigations on inlet and discharge orifices reported in this paper were planned to extend the work of Stach to a larger duct size and to the conditions of air flow met in fan testing. Another object has been to simplify the constructions used by Stach and recommended in the V.D.I. rules.

In the effort to simplify the construction, it was decided to use one measuring tap and, for discharge orifices, to locate this with its center line 1 in. upstream from the orifice-plate surface. This location is known as a *flange tap*.[10] Its use is justified, in pipes or ducts of fair size, by the fact that no differences are observable between a flange-tap pressure reading and a corner-tap reading until the ratio of orifice area to duct area becomes quite large. The precise limits for permissible use of flange taps have not been determined but flange taps are found to be satisfactory in a duct 32.75 in. in diameter with an area ratio $m = 0.8$; and in a duct 27.4 in. in diameter with $m = 0.7$. If there is any question as to the propriety of using a flange tap, it is always possible to substitute a corner tap.

The inside diameter of the flange-tap opening was the subject of investigation. It was found that the readings became more consistent as the inside diameter was increased up to 0.25 in. and this dimension was selected as standard. The tap fitting must, of course, be flush with the inside duct surface.

The Witte investigations[6,7,8,9] indicate no desirable lower limit either to the thickness of the orifice plate or to the length of the cylindrical portion of the orifice. Investigations were made with orifice plates from $^1/_4$ to $^1/_{16}$ in. thick, in all cases with the cylindrical portion $^1/_{16}$ in. long in the direction of flow. No change in the value of the coefficients could be detected as the thickness was altered. In view of their cheapness, lightness, and ease of machining, the thinner plates were selected as standard. These were actually $^1/_{16}$ in. to $^3/_{32}$ in. thick so that there was no need to chamfer the orifices on the downstream side. The orifices were cut in a circular shear which has the effect of rounding the edge of the orifice. It is essential that, on the approach side, the orifice bore should form a sharp corner with the orifice plate. For this reason, the projecting edge of the orifice is filed away and this edge must always be used as the upstream edge. Exact circularity of the orifice is not important when the diameter is large; the orifice area in an orifice 30 in. in diameter is affected negligibly by an out-of-roundness of as much as $^1/_8$ in. The orifice plate must be so stiff that it does not vibrate under any condition of air flow.

To sum up, the conditions finally selected for a discharge orifice for fan use are:

1 An orifice plate $^1/_{16}$ in. to $^3/_{32}$ in. thick.

2 An orifice with sharp corners on the approach side and without chamfer on the discharge side.

3 A flange tap with an inside diameter of $^1/_4$ in. located 1 in. upstream from the orifice plate, except for very large area ratios or quite small ducts, when corner taps are to be substituted.

Investigation of Discharge Orifices

A calibrated well-rounded nozzle is the nearest approach to a primary standard for the measurement of large quantities of air. In the test arrangements for determining the coefficients of discharge orifices, it would have been desirable to have the air under measurement pass successively through a well-rounded nozzle and through the discharge orifice. With ducts up to nearly 60 in. in diameter, such an arrangement involves a considerable length of duct and was too costly. In its place pitot-tube traverses were substituted for the nozzle, the accuracy of the pitot-tube traverses being first demonstrated by a series of preliminary runs in which the air, after being measured by a pitot-tube traverse, was discharged through a well-rounded nozzle. The

[10] "Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air," by H. S. Bean, E. Buckingham, and P. S. Murphy, *Journal of Research*, U. S. Bureau of Standards, vol. 2, January-June, 1929, Research Paper No. 49, p. 561.

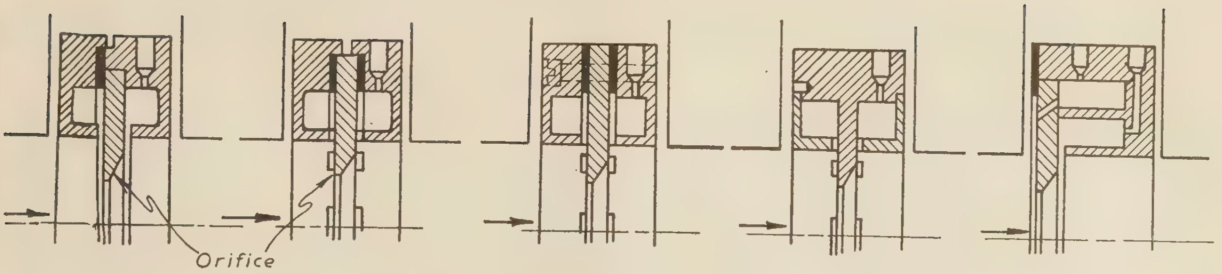

Fig. 2 Ring Piezometers Prescribed in V.D.I. Rules

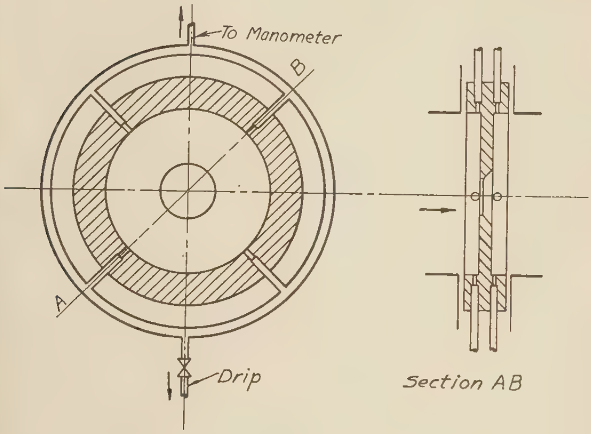

Fig. 3 Orifice With Corner Taps of Drilled Holes Connected Externally

pitot tube used was of the standard design proposed by the author in another paper.[3] The other proposals in the same paper[3] were also followed.

These preliminary runs fully justified the adoption of a pitot-tube traverse as a secondary standard under the conditions of these tests. The differences between the pitot-tube volumes and nozzle volumes were not greater than 0.1 per cent in the majority of tests made. The maximum discrepancies were 0.9 per cent in one case and 0.5 per cent in another.

In all, 170 determinations were made of the discharge-orifice coefficients but, of these, the majority were for the purpose of ascertaining the influence on the coefficient of such factors as orifice-plate thickness, location of pressure taps, and diameter of pressure taps. After these construction details were settled, determinations of discharge coefficients were made on three ducts with diameters of 32.75, 40.4, and 58.5 in., respectively. All

TABLE 1 ORIFICE COEFFICIENTS RECOMMENDED IN V.D.I. RULES [a]

Orifice-area ratio m	Discharge coefficient
0.05	0.598
0.10	0.602
0.15	0.608
0.20	0.615
0.25	0.624
0.30	0.634
0.35	0.646
0.40	0.661
0.45	0.677
0.50	0.696
0.55	0.717
0.60	0.742
0.65	0.770
0.70	0.804
0.75	0.845
0.80	0.900

[a] Coefficients for orifice-area ratios m greater than 0.7 are those found by Witte.[9] The V.D.I. rules give coefficients only for values of m up to 0.7.

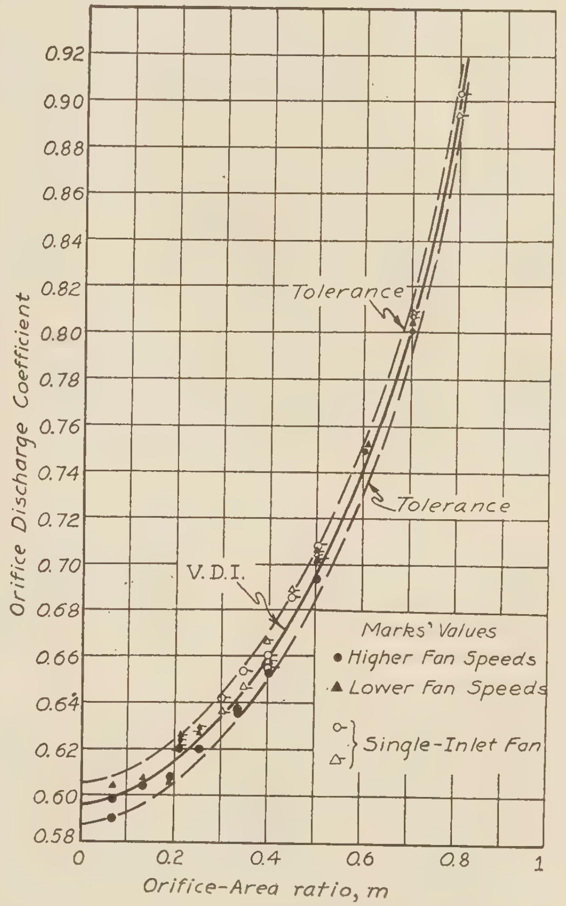

Fig. 4 Discharge Coefficients for Square-Edged Orifices Recommended in V.D.I. Rules Compared With Those Determined by the Author

ducts were at least 10 diameters in length and pitot-tube traverses were made at a distance of 7.5 diameters from the entrance end.

The extreme range of orifice-area ratios investigated was from 0.066 to 0.796, but the range was different for each of the three duct diameters. For each orifice-area ratio and duct diameter, determinations were made at two air speeds, the lower speed being high enough to give dependable pitot-tube indications. The results of these determinations are shown in Fig. 4. The coefficients recommended in the V.D.I. rules are given in Table 1 and are plotted in Fig. 4 together with the coefficient tolerances recommended in the V.D.I. rules. The coefficients for values

of orifice-area ratios m greater than 0.7 as given in Table 1 and plotted in Fig. 4 are those found by Witte.[9] The coefficients recommended in the V.D.I. rules are for orifice-area ratios up to 0.7.

Fig. 4 shows that the coefficients found by the author fall within the V.D.I. tolerance limits. The triangular and circular points at any one orifice-area ratio represent different air speeds and it appears that there is no constant direction of change of discharge coefficients with air speed. Some of the points were obtained with air discharging from a single-inlet fan, others from a double-inlet fan. The discharge coefficients for the single-inlet fan average slightly above those from the double-inlet fan but the magnitude of this difference is insignificant in view of the tolerance allowed.

It may appear strange that a tolerance as high as ± 1.5 per cent is necessary for the discharge coefficient of an orifice. The tolerance proposed by Stach[4] was ± 0.8 per cent. In the author's tests, in order to come within the tolerance of ± 1.5 per cent, it was found necessary to refine the pitot-tube differential-pressure measurements by the use of a Whalen gage[11] and also to measure the drop in orifice pressure with a hook gage. The ordinary inclined-tube manometer was not accurate enough for this work. The variation shown in the ascertained values of the coefficients is believed due to variation in the pattern of the velocity distribution of the air across the duct. The existence of this variation is clearly shown in the pitot-tube traverses.

The coefficients c of Fig. 4 and Table 1 include the approach factor. If the approach factor is given in the equation, and is not included in the coefficient, the coefficient becomes $c' = c\sqrt{1 - m^2}$. Values of c' corresponding to the coefficients of Table 1 are as follows:

m	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
c'	0.599	0.603	0.607	0.610	0.607	0.601	0.586	0.54

It will be seen that these values do not depart much from 0.60 until the orifice-area ratio m exceeds 0.6. This is the value of the coefficient for inlet orifices which have zero velocity of approach. The foregoing relationship is rational, since with small orifice-area ratios the condition of flow at a discharge orifice approximates that at an inlet orifice except for the velocity of approach.

Inlet Orifices

Discharge coefficients for flow of air through thin-plate inlet orifices were determined by Ebaugh and Whitfield[12] for a duct 27.25 in. in diameter and for pressure differentials of 0.1 to 2.0 in. of water. They found that a static-pressure tap located 0.4 of the duct diameter downstream from the orifice plate will measure the pressure differential with an accuracy within 0.5 per cent for orifice-area ratios m ranging from 0.2 to 0.8. They found a coefficient of discharge of 0.601 for orifice-area ratios from 0.2 to 1.00 and for pressure differentials as low as 0.051 in. of water.

Stach,[4] using a thick-plate orifice, corner taps as shown in Fig. 2, and a duct 10 in. in diameter, finds a coefficient of 0.60 with the following tolerances:

Orifice-area ratio m	0.25	0.36	0.49–0.69
Tolerances, per cent	+1.3	± 1.0	± 0.5

The V.D.I. rules prescribe corner taps and give a discharge coefficient of 0.60 with a tolerance of ± 1.5. They also state that the flat, unobstructed diameter of the orifice plate should be at least 1.5 times the diameter of the orifice. (It may be noted that no such ratio existed for the case of $m = 1.00$ in the investigations of Ebaugh and Whitfield[12] but that the coefficient was apparently unaffected thereby.)

The agreement between the coefficients obtained by Stach,[4] using the corner taps, and by Ebaugh and Whitfield,[12] using pressure taps located 40 per cent of the duct diameter downstream from the orifice, becomes rational when we examine the static-pressure variation along the duct. In Fig. 5 of the paper by Ebaugh and Whitfield[12] are plotted curves showing the static pressures at the corner tap and at a series of taps downstream from the orifice. It appears that with a duct 22.7 in. in diameter and with orifice-area ratios up to 0.65, the static pressure changes very slightly from the corner tap to a distance of 40 per cent of the duct diameter. A justifiable inference from the curves is that the same conclusion would be true for even larger orifice-area ratios up to some undetermined limit but probably up to at least $m = 0.80$.

The author has made a number of determinations of discharge coefficients of inlet orifices, using thin-plate orifices and the same location for pressure taps that was used by Ebaugh and Whitfield,[12] namely, 40 per cent of the duct diameter downstream from the orifice plate. With a 30-in. duct, using eight orifices with orifice-area ratios ranging from 0.09 to 0.95, the discharge coefficient had a mean value of 0.601, which chances to be identical with the Ebaugh and Whitfield value. The extreme range in values was from 0.597 to 0.606, or a tolerance from the mean value of 0.8 per cent.

The evidence available would seem to indicate that a square-edged inlet orifice is a very reliable device for measuring air and that the discharge coefficient is constant throughout a considerable range of operating conditions.

Reynolds-Number Limit

The discharge coefficients, both for inlet and discharge orifices, given in this paper, apply to turbulent flow and should be used only when the Reynolds number exceeds certain limits. If the Reynolds number is stated in the form $R = DV\rho/\mu$ where D is the duct diameter and V is the velocity of flow in the duct, the limiting values of R for discharge orifices given in the V.D.I. rules are:

m	0.2	0.3	0.4	0.5	0.6	0.7
$R \times 10^{-4}$	5.0	7.5	10.0	12.5	15.0	17.5

According to Stach[4] the values are:

m	0.25	0.36	0.49	0.64
$R \times 10^{-4}$	4.50	5.50	7.50	18.30

The limits are not very definite and the coefficient does not change rapidly until the value of R is considerably below the limit, so that the agreement between the V.D.I. values and the Stach values just given may be regarded as satisfactory. The value of the coefficient increases when R falls below the stated limits but actual values have not been determined for inlet or discharge orifices.

The stated limiting values of R are normally exceeded in fan testing. For example, using air of standard density (0.075 lb per cu ft) and a temperature of 70 F, $\rho/\mu = 6.05 \times 10^3$ in ft-lb-sec units. Assuming a duct of 1 ft diameter and velocity of 10 fps in the duct, $R = 6.05 \times 10^4$. This value would probably be above the limiting value, since the low velocity assumed would normally correspond to a low orifice-area ratio. The conditions here assumed give as low a value of R as is likely to occur in fan testing. Any doubtful case should be calculated.

[11] "Investigation of Warm-Air Furnaces and Heating Systems," by A. C. Willard, A. P. Kratz, and V. S. Day, University of Illinois Engineering Experiment Station Bulletin No. 120, University of Illinois, Urbana, Ill., March, 1921.

[12] "The Intake Orifice and a Proposed Method for Testing Exhaust Fans," by N. C. Ebaugh and R. Whitfield, Trans. A.S.M.E., vol. 56, 1934, paper PTC-56-3, p. 903.

For inlet orifices, the limiting Reynolds number is given both by Stach[4] and in the V.D.I. rules[5] as 5.5×10^4.

Other Factors

The V.D.I. rules[5] are intended to apply to all single-phase fluids and to orifices as small as 0.1 in. in diameter. They include a series of corrections to be applied to the discharge coefficients. These corrections can in general be disregarded under the conditions met in fan testing. The correction for roughness of the interior surface of the duct is negligible for duct diameters in excess of 6 in. Similarly, the correction for departure from perfection of the square edge of the orifice becomes negligible for a 6-in. duct and an orifice-area ratio of 0.2.

The discussion up to this point has assumed that air acts as a nonexpansible fluid. This assumption, while physically unjustifiable, leads to no appreciable error as long as the pressure drop through the orifice amounts to only a few inches of water. In the A.S.M.E. test codes, a fan is defined as having a maximum pressure rise of 1 lb per sq in.; above that limit the apparatus is called a compressor. Considering this upper limit and assuming that the whole of the pressure change occurs at the measuring orifice during the testing of the fan, the pressure differential in the orifice will be about $^1/_{15}$ of the absolute pressure of the delivered air. The neglect of the expansibility of the air will then necessitate the use of the following correction factors:

m	0.3	0.5	0.6	0.7
Correction factor	0.977	0.975	0.973	0.970

As the correction is practically in direct proportion to the pressure differential in the range of pressure differentials under consideration, it follows that with $m = 0.7$ and a pressure differential of 0.2 lb per sq in., or 5 in. of water approximately, the correction factor becomes 0.994. The correction factor, while negligible for most fans, may be important when the pressure differential through the measuring orifice is high.

Use of Impact Tube

The question was raised by the author in a previous paper[3] as to the feasibility of using an impact tube to determine the flow through an orifice, in the same manner in which it is so successfully used with discharge nozzles. It was thought that this might simplify construction and eliminate the question as to the desirable location of the pressure tap. This matter has been investigated by the author. It appears that in fan discharge ducts the flow pattern of the stream discharging from an orifice is too variable to justify the use of an impact reading taken at any one specified location. For example, with large orifices it is often true that the impact reading at the center of the stream is lower than at either side of the center instead of being a maximum as found in the traverse of the discharge from a well-rounded nozzle.

Summary

The author believes that the investigations herein reported, supporting and extending the work of earlier investigators, can safely be accepted as the basis of a greatly simplified and more accurate procedure in the testing of fans and blowers. In a fan or blower provided with either an inlet or discharge duct, the customary pitot-tube traverse can be replaced by a single observation at an inlet or discharge orifice.

Acknowledgment

The author desires to acknowledge his indebtedness to Randolph Ashton, graduate student, and Howard Emmons, assistant, at the Harvard Engineering School for the skill and patience with which they have carried out the extensive series of observations on which this paper is based.

Undercooling in Steam Nozzles

By J. T. RETTALIATA,[1] MILWAUKEE, WIS.

This paper deals primarily with the effect of wall roughness on the flow of steam in nozzles; and secondarily with drop growth and the occurrence of initial condensation. It was found that roughness of the nozzle walls caused a retardation of steam flow and resulted in the Wilson line's occurring at the 3.2 per cent moisture line on the Mollier diagram instead of at the 3.7 per cent moisture line as was found for a nozzle with smooth walls. In a specially designed nozzle drop growth was accomplished as evidenced by the complete visible spectrum which resulted from the scattering of light by the expanding steam in this nozzle. Photographs of the condensation region show that condensation is not occurring at any point previous to where the eye sees it. These researches indicate that the location of the Wilson line on the Mollier diagram depends entirely upon the rate of change of velocity of the steam, and, therefore, cannot be fixed for all conditions. In accordance with this fact it is suggested that the term "Wilson line" is a misnomer and should be replaced by the more appropriate one of "Wilson zone."

THERE HAS been a vast amount of experimental work done on the flow of steam in nozzles. However, the field has not been fully covered and there are many problems yet to be solved. The research discussed in this paper was undertaken to investigate the influence of wall roughness on the flow of steam in nozzles; to ascertain whether drops could be made to grow in a specially designed nozzle; and to determine optically whether initial condensation was actually occurring in a nozzle at the same point where the eye detected it.

In 1932 and 1933, J. I. Yellott (1)[2] investigated drop size and supersaturation by studying the flow of low-pressure steam through nozzles similar to those used in turbines. The original object of the investigation was to study the condensation of steam in an effort to discover how and when initial condensation occurs. It was desired to locate by experimental methods the Wilson line, which on the Mollier diagram represents the condition at which initial condensation starts when steam expands in the supersaturated or undercooled state.

The problem was attacked with the aid of light rays and applications of the laws of optics. If the initial condensation point could be seen, its pressure could be measured with a search tube, and in this manner some positive knowledge of supersaturation could be obtained. Also, the size of any droplets large enough to be seen could be determined to a reasonable degree of accuracy by means of the wave length of light reflected or scattered by them, according to Rayleigh's principles.

Yellott's experiments formed a starting point for the author's investigations and much of Yellott's equipment was available for the researches described in this paper. The apparatus used is the same as that shown in Fig. 3 of Yellott's article (1) except that the pressure was measured at point H instead of at G. This was done so as to obtain the pressure of the steam at the same point where its temperature was measured.

Experimental Work

Nozzle A shown in Fig. 1 was the first to be tested. This was a smooth-walled nozzle similar to Yellott's nozzle No. 1. A pressure traverse was made on nozzle A by moving the search tube along the nozzle axis and measuring the pressures by means of a Bourdon gage and mercury column. These readings were taken every 0.2 in. along the nozzle axis, except in the condensation region where they were taken every 0.05 in. so as to get a more accurate conception of the pressure variation in this zone.

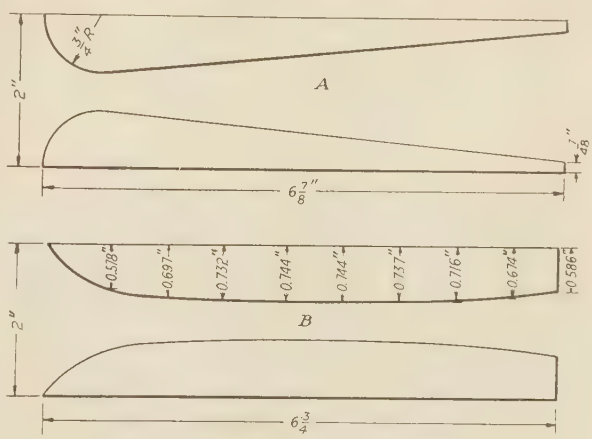

Fig. 1 Dimensions of Nozzle Blocks

The condensation region could be immediately detected by the occurrence of a dense bluish fog caused when the light was scattered by the moisture particles.

The pressure traverses were run using inlet pressures of 44.7, 49.7, 54.7, and 59.7 lb per sq in. abs, and corresponding back pressures of 32.8, 34.8, 37.6, and 39.7 lb per sq in. abs, respectively. Because of the better analysis that can be made of the results when they are in curve form, the data on the various nozzles are shown as such in Figs. 2 to 5, inclusive.

The dashed curves represent the smooth nozzle. Analyzing the curve in Fig. 2, the steam expands regularly until a pressure of 19.6 lb per sq in. abs is reached and then there is an abrupt rise in pressure. After this, the steam expands regularly as before. This abrupt rise in pressure occurs exactly at the point of initial condensation. This could be seen by turning on the arc light and noticing that the search-tube pressure holes were exactly at the curved line denoting condensation when the pressure rise was recorded. The reason for this pressure rise at initial condensation will be explained later.

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[2] Numbers in parentheses refer to Bibliography at the end of the paper.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

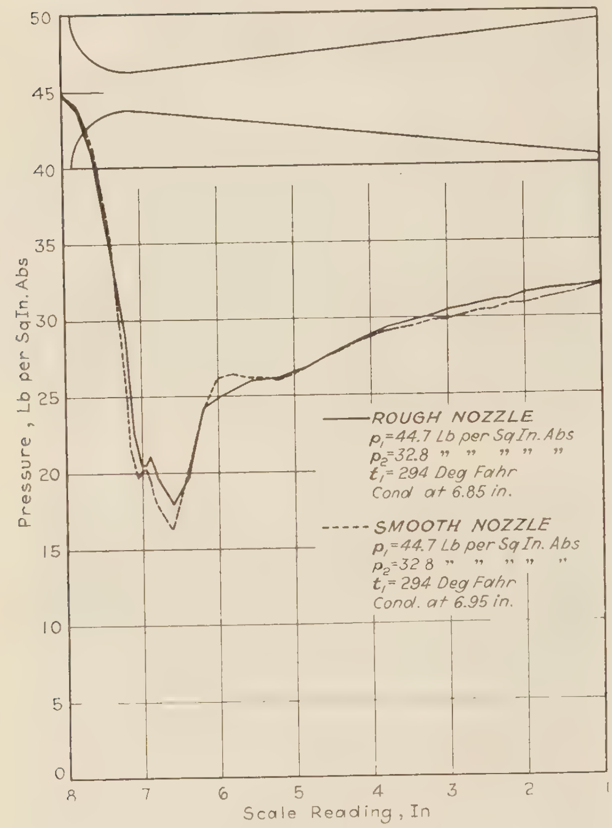

Fig. 2

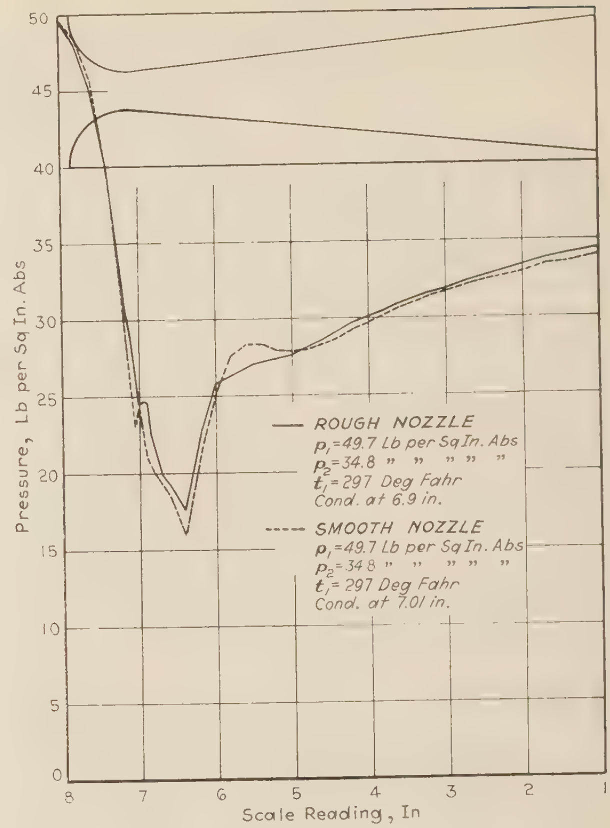

Fig. 3

The expansion continues past initial condensation and reaches its lowest point at 16.2 lb per sq in. abs. At this point recompression starts and continues until an absolute pressure of 26.4 lb per sq in. is reached. Next a slight drop in pressure to 26 lb per sq in. abs occurs, and after this the pressure increases to the back pressure. This curve was typical of every pressure traverse that was made on the smooth nozzle.

The type of roughness that was adopted was obtained by making indentations on the surface of the nozzle blocks with a cape chisel. This type was selected as nearest to that which would be found in service where corrosion forms pit holes. Fig. 6 shows a comparison of the rough-walled and the smooth-walled nozzles.

In the pressure traverses made on the rough nozzle, the same inlet and back pressures were established as with the smooth nozzle. The conditions were kept identical for comparative purposes. The results are shown in Figs. 2 to 5, inclusive, and were plotted on the same axes so they could be compared readily. The solid line represents the rough nozzle.

Again referring to Fig. 2, the steam expands regularly from the inlet condition in the rough nozzle until a pressure of 20.5 lb per sq in. abs is reached, and then the abrupt pressure rise at initial condensation is encountered as in the case of the smooth nozzle. This pressure rise, however, occurs at a higher pressure in the rough nozzle than in the smooth nozzle. This difference will be explained later. After initial condensation the steam expands regularly as before until it reaches its lowest pressure at 17.9 lb per sq in. abs. This low-point pressure is higher than the low-point pressure that occurred in the smooth nozzle.

Comparing the two expansion parts of the curves, the steam expanding in the smooth nozzle has a higher pressure than the steam in the rough nozzle until a pressure of 33.5 lb per sq in. abs is reached, at which point they intersect. After this intersection, the steam in the smooth nozzle expands more rapidly and to a lower pressure than the steam in the rough nozzle. These expansion curves are in direct agreement with those of Mellanby and Third (2) in their work on rough- and smooth-walled converging nozzles. From the appearance of these curves it is evident that the roughness of the walls retards the expansion of the steam in the converging portion of a nozzle.

Undercooling will now be discussed to explain the higher condensation pressure occurring in the rough nozzle. The condition of the steam at the initial condensation point is determined by assuming the expansion to the Wilson line to be isentropic. The condensation point can be found on the Mollier diagram by following the entropy line on which the initial condition is located. The intersection of this entropy line and the line representing the pressure at which condensation was observed will be the initial condensation point. This expansion is shown in Fig. 7.

The point A represents the initial steam conditions. The constant-entropy line from A crosses the saturation line at B and this would be the condensation point if the expansion were slow enough to proceed under conditions of equilibrium. However, in the case of the smooth nozzle, the steam is expanding so rapidly that condensation is not allowed sufficient time to start at B, and undercooling results. Consequently, without equilibrium present, the condensation point for the smooth nozzle is at D instead of B. The pressure corresponding to point D is represented by the line P_d which is the condensation pressure in the smooth nozzle. The expansion in the rough nozzle is not sufficiently slow to have equilibrium conditions present, resulting in condensation at B, nor is it rapid enough (due to roughened walls which decrease the steam velocity) to permit condensation to

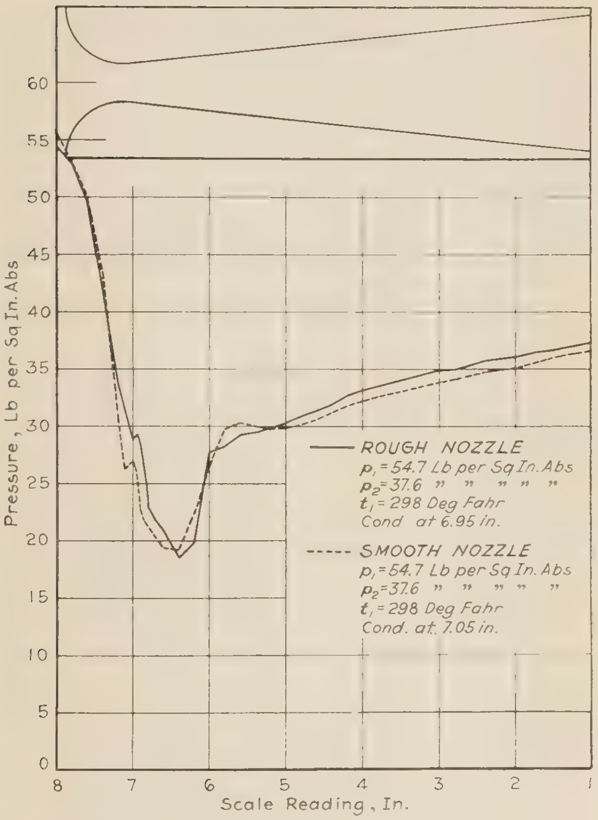

Fig. 4

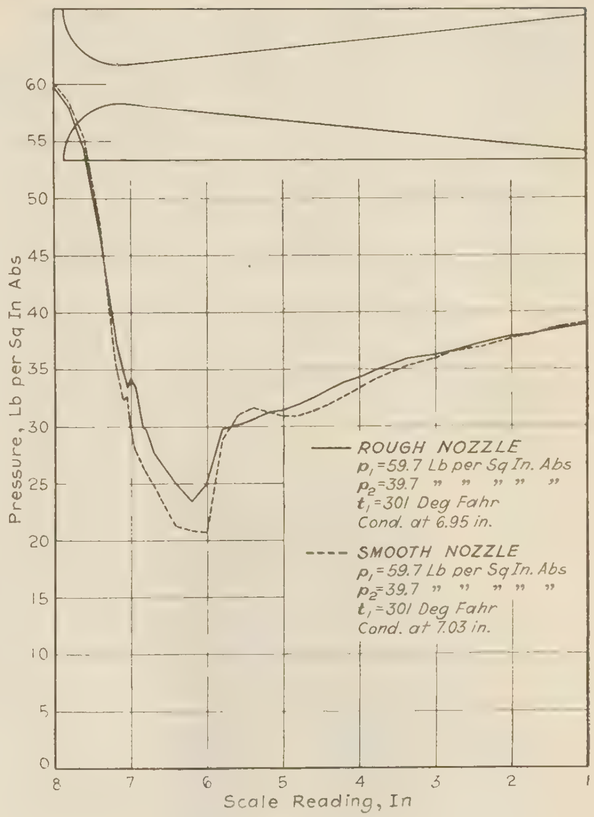

Fig. 5

occur at *D*, as in the case of the smooth nozzle. Hence, some point *C*, between *B* and *D*, will be the condensation point for the rough nozzle. Therefore some pressure P_c at *C* will represent the condensation pressure in the rough nozzle and is higher than P_d, the condensation pressure in the smooth nozzle.

An explanation of the occurrence of the abrupt pressure rise at initial condensation can be given as follows: As the steam passes from the undercooled state, condensation begins and the condensed water droplets throw their latent heat back into the vapor portion of the steam mass. Essentially, the volume occupied by the steam at the termination of the undercooled state is momentarily the same as that at the beginning of condensation. Therefore, the latent heat liberated by the water droplets heats the steam at practically constant volume which raises the pressure. J. H. Keenan (3) in his discussion of Yellott's paper (1) presents a mathematical treatment of this abrupt pressure rise at initial condensation.

Referring again to Fig. 2, after the steam in the rough nozzle has expanded to its lowest pressure, it recompresses rather quickly at first and then there is a gradual rise in pressure until the back pressure is reached. It is to be noted, however, that in no case does a slight drop in pressure occur after recompression as in the smooth nozzle. This is also shown in Figs. 3, 4, and 5.

This phenomenon is caused by the roughened walls retarding the rate of compression. Therefore, the pressure rise is gradual until the back pressure is reached. In the smooth nozzle, however, due to this lack of retardation, the recompression proceeds with such rapidity that there is an overcompression to a higher pressure, and a resulting fall in pressure in a lower portion of the nozzle.

These secondary recompression effects in a nozzle were also observed by Stodola (4). He interpreted these extraordinary surges of pressure to be the realization of Riemann's (5) theory of compression shock, that is, the rapidly moving steam particles impinge upon a mass of steam giving way too slowly and are thereby compressed to a higher pressure. Experiments carried out later by Stodola show that this recompression always involves a greater or lesser detachment of the steam jet from the nozzle wall.

The general shape of the curves shown in Figs. 2 to 5, inclusive, is the same, in that each shows an overexpansion and then a recompression to the back pressure. When the recompression attained a pressure corresponding to that at which condensation occurred, a dark spot was noticed in the steam flowing through the nozzle which was caused by reevaporation of most of the water droplets. Whether or not an abrupt pressure change would occur at this reevaporation point on the recompression line would depend upon the rate of change of pressure during compression. That is, when the change of pressure is very rapid, evaporation may occur at a higher pressure than that of equilibrium, and a restoration to conditions of equilibrium would be accompanied by a fall in pressure due to the abstraction of latent heat of vaporization. Evaporation occurs under conditions other than those of equilibrium when steam is desuperheated by the injection of water into it. In this process it has been observed that water droplets have continued to persist even in this atmosphere of superheated steam, thus indicating a removal from equilibrium conditions.

The Wilson line is located on the Mollier diagram by the method described in Yellott's paper (1) which is equivalent to assuming an isentropic expansion from the initial condition to the condensation pressure (represented by extending the superheat pressure lines into the wet region), and then an isenthalpic change

from these extended pressure lines to the conventional pressure lines in the wet region.

The rough nozzle affects undercooling of the steam by changing the location of the Wilson line that was found for the smooth nozzle. The data that were used to locate the Wilson line are the same that were plotted in the pressure traverses in Figs. 2 to 5 and are shown in Table 1. Using these data for the rough nozzle,

TABLE 1 RESULTS OF TESTS ON ROUGH AND SMOOTH NOZZLES

Point	Initial pressure, lb per sq in. abs	Initial temp, F	Condensation pressure, lb per sq in. Rough nozzle	Smooth nozzle
A	59.7	301	33.3	32.2
B	54.7	298	28.8	26.1
C	49.7	297	24.5	22.9
D	44.7	294	20.5	19.6

the Wilson line was found to be located near the 3.2 per cent moisture line on the Mollier diagram as shown in Fig. 8, wherein points A, B, C, and D represent initial conditions and points A',

Fig. 6 Comparison of Rough and Smooth Nozzles

B', C', and D' represent conditions at condensation for the rough nozzle. Points D and D' refer to Fig. 2; points C and C' refer to Fig. 3, etc. The Wilson line for the smooth nozzle was found to lie around the 3.7 per cent moisture line as shown in Fig. 9.

This difference in location of the Wilson line can be explained very effectively when consideration is given to the time element during expansion. Goodenough (6) has pointed out that "a change of state with equilibrium constantly maintained is an ideal process which is never realized, and which if realized would require infinite time. Equilibrium represents a static, dead condition. Any movement from an equilibrium state must be preceded by a disturbance of equilibrium, and the more rapid the change, the greater is the departure from equilibrium." A sample calculation will be performed to determine the time interval which elapses after the expanding steam crosses the saturation until it reaches the Wilson line on the Mollier diagram. For comparative purposes this time interval will be determined for the smooth and the rough nozzle. The conditions depicted by Fig. 5 will be used in the calculations.

The velocity of the expanding steam at any point is given by the equation

$$V = 223.8\sqrt{\epsilon H} \qquad [1]$$

where, V = velocity, fps; ϵ = nozzle efficiency, expressed as a decimal; and H = isentropic heat drop from an initial condition to the selected point, Btu.

In particular, the velocity of the steam at the saturation line may be expressed by

$$V_s = 223.8\sqrt{\epsilon H_s}$$

where, H_s = isentropic heat drop from the initial condition to the saturation line, Btu.

Likewise the velocity of the steam at condensation may be obtained from

$$V_c = 223.8\sqrt{\epsilon H_c}$$

where, H_c = isentropic heat drop from the initial condition to the condensation point, Btu.

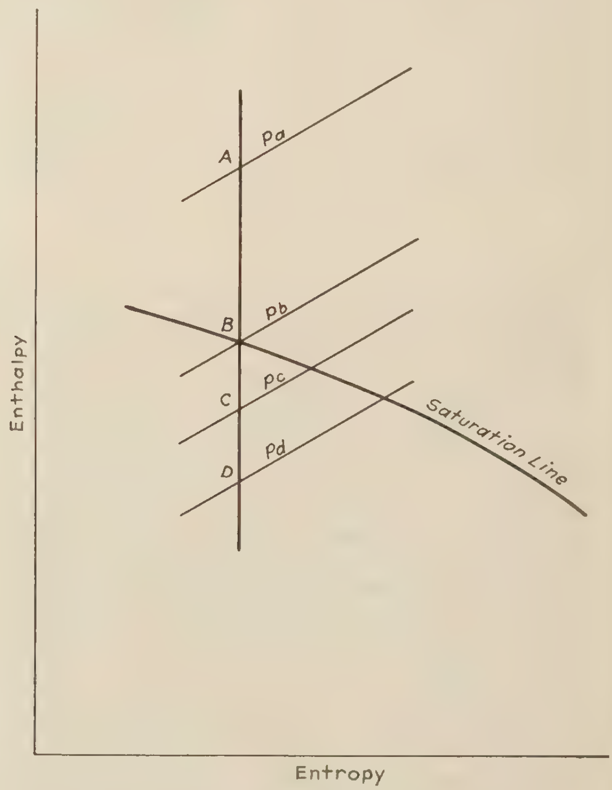

Fig. 7

The average velocity of the steam from the saturation line to the condensation (or Wilson) line may be expressed by the relation

$$V_{av} = \frac{1}{H_c - H_s}\int_{H_s}^{H_c} 223.8\sqrt{\epsilon H}\, dH$$
$$= \frac{223.8\sqrt{\epsilon}}{H_c - H_s}\int_{H_s}^{H_c} \sqrt{H}\, dH$$

$$= \frac{223.8\sqrt{\epsilon}}{H_c - H_s}\,\frac{2}{3}\left[(H_c)^{3/2} - (H_s)^{3/2}\right]$$

$$= \frac{149.2\sqrt{\epsilon}}{H_c - H_s}\left[(H_c)^{3/2} - (H_s)^{3/2}\right] \qquad [2]$$

The conditions given by Fig. 5 for the smooth nozzle are: $t_1 = 301$ F; $p_1 = 59.7$ lb per sq in. abs; $p_s = 55.0$ lb per sq in. abs; $p_c = 32.2$ lb per sq in. abs; $\epsilon = 99.5$ per cent assumed nozzle efficiency; and $v_1 = 7.31$ cu ft per lb.

The specific volume at condensation v_c, can be calculated

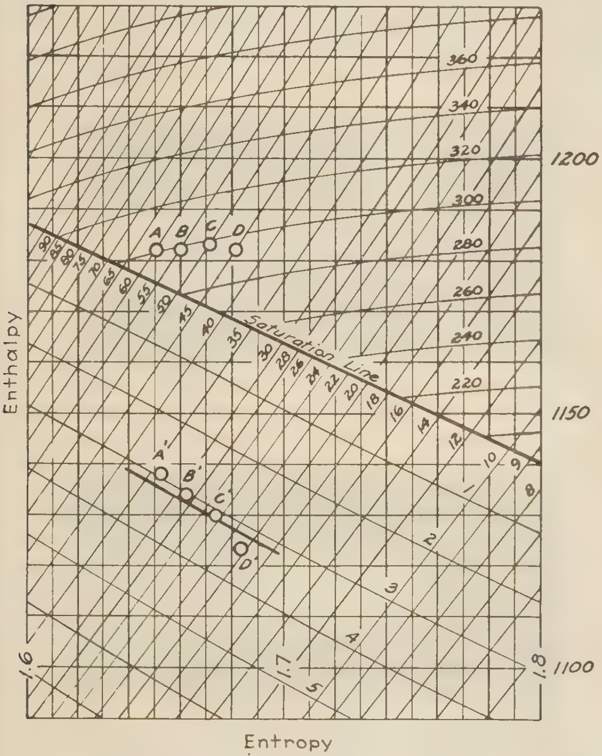

FIG. 8 MOLLIER DIAGRAM SHOWING WILSON LINE LOCATED AT 3.2 PER CENT MOISTURE LINE FOR THE ROUGH NOZZLE

from the perfect gas relationship pv^n = constant, where $n = 1.3$. This gives

$$v_c = 7.31\left[\frac{59.7}{32.2}\right]^{0.768} = 11.75 \text{ cu ft per lb}$$

The isentropic change in enthalpy from the initial pressure to the condensation pressure for undercooled conditions is given by the equation

$$H_c \doteq \frac{144}{778} \times \frac{n}{n-1}\left[p_1v_1 - p_cv_c\right]$$

from which is obtained

$$H_c = \frac{144}{778} \times \frac{1.3}{1.3 - 1}\left[59.7 \times 7.31 - 32.2 \times 11.75\right]$$

$$= 46.95 \text{ Btu}$$

The isentropic change in enthalpy H_s, from the initial pressure to the saturation pressure is found to be 6.36 Btu from the Mollier diagram.

Substituting the values for H_c, H_s, and ϵ in Equation [2], the average velocity from the saturation line to the Wilson line is

$$V_{\text{avg}} = \frac{149.2\sqrt{0.995}}{(46.95 - 6.36)}\left[(46.95)^{3/2} - (6.36)^{3/2}\right]$$

$$= 1121 \text{ fps}$$

The distance D traversed by the steam in the smooth nozzle during the interval of expansion from the saturation line to the Wilson line is found from Fig. 5 to be 0.5 in.

Therefore the time interval x which elapses after the expanding

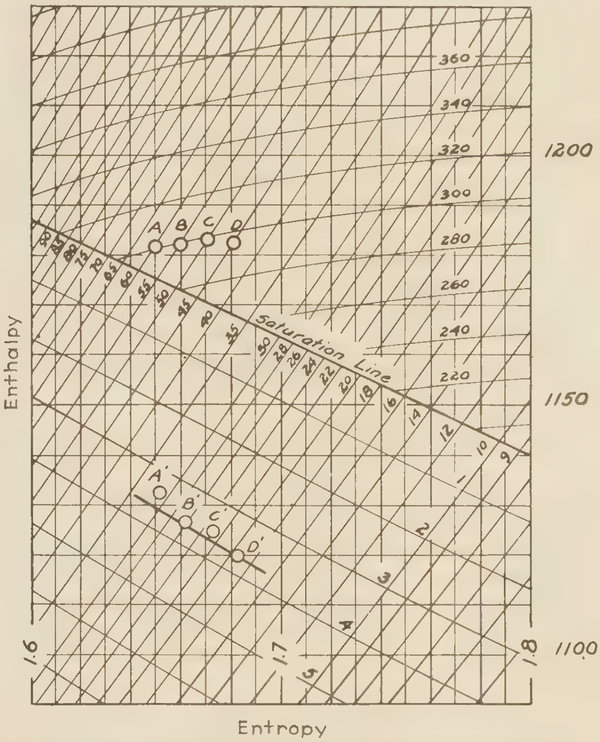

FIG. 9 MOLLIER DIAGRAM SHOWING WILSON LINE LOCATED AT 3.7 PER CENT MOISTURE LINE FOR THE SMOOTH NOZZLE

steam crosses the saturation line until it reaches the Wilson line on the Mollier diagram is given by

$$x = D/V_{\text{avg}} = 0.50/(12 \times 1121) = 3.71 \times 10^{-5} \text{ sec}$$

By a similar reasoning the time interval for the rough nozzle, with an assumed nozzle efficiency of 97 per cent, is found to be 4.68×10^{-5} sec.

By a comparison of the time interval and also referring to Goodenough's statement (6), it can be seen that the expansion in the rough nozzle approaches infinite time (and hence equilibrium) closer than the expansion in the smooth nozzle. Therefore, the Wilson line for the rough nozzle will lie closer to the saturation line (which represents conditions of equilibrium) than the Wilson line for the smooth nozzle.

The size of the moisture droplets in the rough nozzle was calculated from the von Helmholtz (7) equation to be 7.37×10^{-8} cm in radius, and those in the smooth nozzle to be 6.35×10^{-8} cm in radius. It is evident that this retarded expansion allows more time for droplet growth.

THEORETICAL NOZZLE

In the case of each nozzle previously tested, the expansions, as

can be seen from the pressure traverses, have not been uniform. Therefore, a theoretical nozzle was designed to give a uniform pressure drop per inch of length along the nozzle axis. A sketch of this nozzle is shown as nozzle *B* in Fig. 1. The pressure traverse as designed would give a straight-line expansion curve. The nozzle was designed for inlet conditions of 55 lb per sq in. abs, 305 F, a back pressure of 10 lb per sq in. abs, and 100 per cent efficiency.

The results of the pressure traverses made on this nozzle are shown in curve form in Fig. 10. The straight dashed line repre-

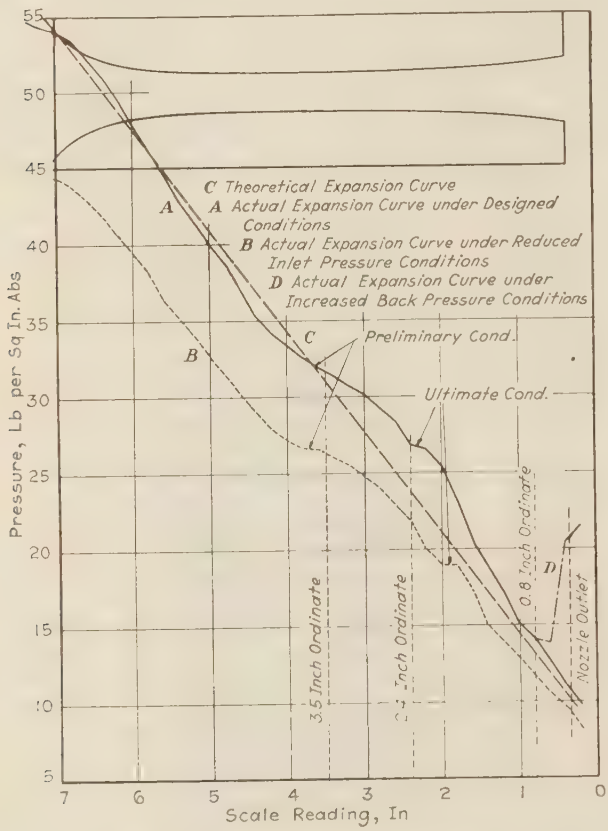

FIG. 10 RESULTS OF TESTS ON NOZZLE *B*

sents the pressure traverse for which the nozzle was designed. The solid line *A* represents the actual pressure traverse obtained when the nozzle was tested under the conditions for which it was designed. On curve *A*, the pressure is continuously falling as the steam expands in the nozzle, that is, at no point does a recompression take place as was the case with the other nozzles tested. When the steam flow in this nozzle was viewed with the carbon arc turned on, a slight bluish haze, denoting preliminary condensation, was noticed at a point in the nozzle corresponding to the 3.5-in. ordinate on the scale. At the 2.4-in. ordinate the familiar darker blue color denoting ultimate condensation was observed. Referring to these two ordinates in Fig. 10, there is a leveling off of the curve which is due to the liberation of the latent heat of condensation (which occurs at these points) tending to raise the pressure.

The nozzle was next tested under inlet conditions for which it was designed, but with a back pressure of 21.5 lb per sq in. abs. Curve *A* shows the expansion to be the same as before until the 0.8-in. ordinate is reached where the pressure begins its rise to the back pressure as shown by the dot-and-dash line *D*.

The nozzle was then tested under a set of conditions entirely different from those for which it was designed, namely, inlet conditions of 44.7 lb per sq in. abs, 280 F, and a back pressure of 8.3 lb per sq in. abs. The results of this test are shown by curve *B* in Fig. 10. The expansion is similar to that under conditions for which the nozzle was designed as shown by curve *A*.

The most valuable results from the tests of this nozzle were the observations of different colors scattered by the moisture particles in the steam. After the blue color, denoting ultimate condensation, there came yellow, orange, and red colors giving the rest of the complete visible spectrum. The size of the moisture particles in the steam is a function of the wave length of the color of the light which it scatters. It will be remembered that as the visible spectrum proceeds from the violet to the red, the wave length of the light is increasing. Therefore, the size of the drops of moisture

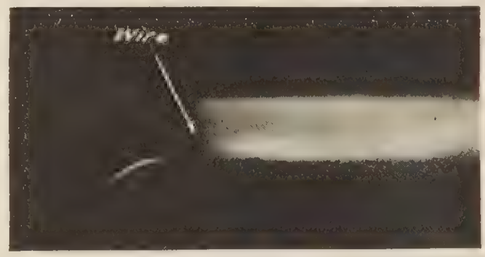

FIG. 11 STEAM FLOW IN NOZZLE *A* PHOTOGRAPHED THROUGH A GREEN FILTER. NOTE THE WIRE TANGENT TO THE CURVE OF INITIAL CONDENSATION

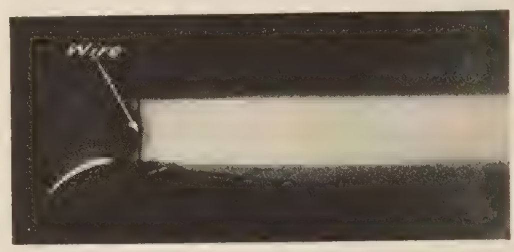

FIG. 12 STEAM FLOW IN NOZZLE *A* PHOTOGRAPHED THROUGH A COBALT GLASS FILTER. THE WIRE IS STILL TANGENT TO THE CURVE OF INITIAL CONDENSATION

in this nozzle is increasing as the steam expands. This could be expected, because the long throat in the nozzle retards the rapidity of the expansion and gives the drops of moisture more time to grow. Hence, in commercial nozzle designs, it would be desirable to eliminate any long throats or parallel sections which encourage drop growth, for it is the large droplets that are most destructive to the low-pressure blading of a steam turbine.

INITIAL CONDENSATION

With every nozzle that was tested during this research, the initial condensation always scattered light of a blue color. The possibility of the initial condensation scattering violet light (of shortest wave length in the visible spectrum), which was not being detected by the eye, suggested itself. This could occur because of the nature of the intensity curve of the violet light and the sensitivity curve of the eye in the violet region. However, the violet light, if present, would sensitize a photographic plate. An experiment was then performed on nozzle *A* shown in Fig. 1. A wire was fastened across the glass cover plate so that when viewed from above by the eye the wire appeared to be tangent to the curved line denoting initial condensation. The purpose of this wire was to locate definitely the position of ultimate conden-

sation as seen by the eye. This arrangement was photographed through a green filter on a hypersensitive panchromatic photographic plate. The green filter was used because, due to the wave length of green light, it would enable a photograph to be taken that would depict the arrangement exactly as the eye saw it. The view obtained is shown in Fig. 11. The same arrangement was next photographed through a cobalt glass filter on an isochromatic photographic plate. This use of the cobalt filter would give a picture that would reveal the violet light if any were present. The result obtained with the cobalt filter is shown in Fig. 12. Figs. 11 and 12 are identical in that the wire is tangent to the curved line of initial condensation in both, indicating that the initial condensation is occurring exactly as the eye sees it.

If any condensation were occurring previous to that seen by the eye it would be indicated in Fig. 12, by the appearance of a white region to the left of the wire.

Figs. 11 and 12 substantiate the location of the initial condensation line as determined from the pressure rise in the pressure traverse at condensation.

Results and Conclusions

The investigation discussed in this paper was intended primarily to determine the effect of wall roughness on the flow of steam in nozzles. Secondary objects were to ascertain whether drops could be made to grow in a specially designed nozzle, and to determine optically whether initial condensation was actually occurring in a nozzle at the same point where the eye detected it.

It was found that wall roughness, causing a retardation of steam flow, made the point of initial condensation occur at a higher pressure and farther downstream than in the smooth-walled nozzle. This caused the Wilson line to occur at the 3.2 per cent moisture line on the Mollier diagram for the rough-walled nozzle, instead of at the 3.7 per cent moisture line as was found for the smooth nozzle. This difference in location of the Wilson line can be explained when consideration is given to the time element during expansion. Calculations revealed that in the case of the rough nozzle the time interval which elapses after the expanding steam crosses the saturation line until it reaches the Wilson line is 4.68×10^{-5} sec, whereas for the smooth nozzle it is 3.71×10^{-5} sec. Hence, more time is allowed for condensation to occur in the rough-walled nozzle which results in the Wilson line lying closer to the conventional saturation line (which represents a condition allowing infinite time for the occurrence of condensation).

The complete visible spectrum was observed in nozzle *B* shown in Fig. 1, indicating a growth of the droplets during the slow expansion of the steam in the nozzle. These results show the desirability of eliminating, in actual turbine design, any long parallel passages that would give the droplets time to grow and result in more rapid deterioration of the low-pressure turbine blading.

The radius of the droplets in the smooth nozzle was found to be 6.35×10^{-8} cm at the beginning of ultimate condensation. The droplet radius in the rough nozzle, however, was 7.37×10^{-8} cm, indicating that the retarded expansion in this nozzle allows the droplet more time to grow.

By employing special filters, photographs of the condensation region show that condensation is not occurring at any point previous to where the eye can detect it.

These researches indicate that the location of the Wilson line on the Mollier diagram cannot be fixed for all conditions, but that initial condensation depends entirely upon the rate of change of velocity of the steam. The Wilson line is located at lower moisture contents up to 4 per cent, the location being dependent on the rate of change of the steam velocity. In accordance with this fact it seems that the term "Wilson line" is a misnomer and should be changed to "Wilson zone" because of its variable location. The term "Wilson line" may, on the other hand, be considered as the lower limit of the condensation zone. The upper limit is indeterminate and depends, as noted previously, upon the time rate of flow of the jet. In the experiments covered by these studies, this upper limit appeared to be at about the 2 per cent normal moisture line on a Mollier diagram. The initial condensation zone on a Mollier diagram, therefore, covers that portion lying between the 2 and 4 per cent normal moisture lines.

Yellott, in more recent determinations not yet published, has found the Wilson line for the smooth nozzle to lie between the 4 and 5 per cent moisture line on the Mollier diagram. This difference between Yellott's and the writer's results for the smooth nozzle can be expected because of the variable time rate of flow existing depending upon the equipment used. It is not likely that any two experimenters would exactly duplicate each other's location of the Wilson line; but their findings should lie within a certain zone.

Acknowledgments

The author wishes to take this opportunity to acknowledge his indebtedness to Prof. A. G. Christie under whose direction this work was performed. He further wishes to express his appreciation to Prof. J. C. Smallwood for his helpful suggestions, and to Prof. A. H. Pfund for aid rendered in the optical phases of the research.

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Turbine Supervisory Instruments and Records

By J. L. ROBERTS[1] and C. D. GREENTREE,[2] SCHENECTADY, NEW YORK

This paper includes a description of a set of instruments for the electrical measurement and detection of such mechanical quantities as shaft eccentricity, bearing vibration, shell expansion, and interference or rubbing of rotating parts. It also contains an analysis of typical, as well as special, starting- and loading-sequence records taken on a 160,000-kw steam and a 20,000-kw mercury-vapor turbine generator, and conclusions which pertain to the adaptability of the instruments to turbine operation.

THE GENERAL Electric Company has developed telemetered recording instruments to meet the demand for protection and supervision, from a remote point, of large turbines. These instruments measure and record the shaft eccentricity, bearing vibration, and shell expansion, and also detect interference or rubbing of rotating parts.

The installation of turbine supervisory instruments on existing central-station turbines has brought to light considerable new and unexpected data pertaining to the mechanical operating characteristics of large turbines. These instruments were originally conceived as vital adjuncts to the remotely operated outdoor power-generation plant. The absence of any large, remotely operated plant, has precluded the testing of these devices under intended conditions, but their operation for the past two years on a 160,000-kw steam-turbine generator set at the Hudson Avenue station of the Brooklyn Edison Company, and on the 20,000-kw outdoor Emmet mercury-vapor turbine-generator set of the New York Power and Light Company at Schenectady, N. Y., has shown their adaptability and value to attended station turbine-generator equipment.

Need for Instruments

To insure safe and successful performance in the starting, stopping, and normal running of turbine generators it has been necessary to rely, to a great extent, on the power of observation of turbine operators and their ability to describe that which took place. Starting cycles and normal running instructions were prepared by the manufacturers for individual units recommending the procedure to follow. The recommendations were based on observations of hundreds of machines in service, and in general have provided safe but probably conservative technique for individual machines.

Based on experience gained with individual units, operators have modified their practice. Unusual conditions, both within and without the turbine, influence the manner in which the turbine should be operated. To obtain the greatest efficiency from any given station, it is necessary that the individual turbine-generator units be capable of being started in the minimum length of time and operated without any danger of impairing their efficiency or mechanical performance.

The instruments provide the operator with an indication and also a permanent record of mechanical performance throughout the starting period and subsequent running time. These permanent records permit all interested individuals to review the past operation of the turbine and plan the future operating method to be employed, instead of having to rely on the opinion of one operator. The instruments are designed for continuous operation and are practically independent of line-voltage changes between the limits of plus or minus 5 per cent.

Eccentricity Recorder

The complete eccentricity-recorder unit consists of the recording instrument and the control unit, the detector coil, mounting yoke, turning-gear switch, shaft ring, and 500-cycle power which is furnished by a small motor-generator set.

The detector coil is mounted on a yoke attached to the turbine foundation, and presented to a ring on the turbine shaft. Any eccentricity of the shaft changes the gap between the coil and the shaft ring which induces variations in the 500-cycle current. This varying current feeds into the control unit, the output of which operates a photoelectric recorder. The recorder measures shaft eccentricity in thousandths of an inch throughout the range from turning-gear speed to synchronous speed. The scale may be adjusted to meet the individual requirements. The reading is independent of shaft shift, that is, of long-time lateral or of vertical displacements or of climbing of the shaft in its bearings.

It is necessary to have two ranges of sensitivity, one for operation at turning-gear speed and the other for all higher speeds. This is controlled automatically by a switch on the turbine, operated by the turning-gear mechanism, so that the correct sensitivity range is automatically obtained without attention from the operator. However, the records are dissimilar and the chart will indicate whether the turbine was on turning-gear or steam operation at any time. By pressing a calibrating button, which supplies a known input to the control unit, the accuracy and maintenance of calibration may be checked when so desired.

This instrument provides accurate knowledge of the shaft eccentricity and makes it possible to start and put the turbine back on the line after a shutdown in a shorter time than formerly, when eccentricity values were inadequately known. Without this knowledge it is necessary to allow ample time, in the interest of safety, to assure the straightness of the shaft, whereas, with it, the condition of the shaft is known at all times and the operator can govern his starting cycle accordingly.

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Contributed by the Power Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held at New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Vibration-Amplitude Recorder

A complete vibration-amplitude recorder unit consists of the recording instrument, amplifier, time switch, push-button station, and the detector units.

A vibration-detector unit is mounted on each of the turbine bearings on which the vibration amplitude is to be measured. These detector units are seismographically mounted electromagnetic devices which transform the mechanical vibrations to voltage variations by means of a coil moving in a permanent-magnet field. These detectors are connected to a vacuum-tube amplifier. The recorder is a standard type of recording voltmeter. Standard full-scale deflection is approximately 0.01 in. and the scale is marked in vibration amplitudes of thousandths of an inch. It is expanded for the small values of vibration amplitude so that an indication of 0.001 in. causes a deflection approximately 40 per cent of full scale. This enables the reading of amplitudes as small as 0.00025 in. The scale may be adjusted to meet individual requirements. The device will record the amplitude of bearing vibration at any frequency from 10 to 30 or 60 cycles per sec for 1800-rpm or 3600-rpm machines.

Fig. 1 Panels for Turbine Supervisory Instruments
(From left to right the panels record eccentricity, vibration amplitude, expansion, and interference.)

The one recording instrument may be used for all of the detectors for one machine. This is accomplished by passing the signal from each detector unit through a telechron motor-driven switch, thus connecting each detector in turn to the amplifier and recorder for a definite time interval. In addition to this motor-driven switch, there is a push-button station which interrupts the normal circuits and permits the reading of the vibration amplitude as picked up by any one of the detectors, regardless of the normal connection through the time switch at any moment.

A calibrating button is provided which will supply a known input to the amplifier, so that the accuracy and maintenance of calibration may be checked at any time.

The continuous and accurate knowledge of the vibration amplitude of several bearings obtained with this instrument provides a much more convenient, sensitive, and reliable check on their operation than the general practice of periodic, portable, hand-operated instrument readings taken by an operator.

Expansion Recorder

A complete expansion recorder consists of a recording instrument and a displacement unit. The displacement unit consists of a voltage-dividing resistor mechanically rotated by the displacement to be measured. The position of the divider arm, expressed in terms of current, determines the reading of the recorder. The remote recorder indicates expansion from cold conditions on a linear scale graduated in tenths of an inch.

This displacement unit was developed primarily to measure relative linear expansion movements between the turbine and its foundation, wherever such measurements are required.

As with the other instruments a continuous reading of the amount and rate of expansion is an improvement over the intermittent checking of the chisel marks on the turbine by an operator.

Interference Detector

The complete interference detector consists of a high-gain amplifier, headphones or loud-speakers, and the detector unit or units. The interference detector was developed to replace the listening rod which, when placed against various parts of the turbine, transmits the internal noises to the ear. This detector operates on the electromagnetic principle, that is, a voltage proportional to the velocity of vibration is generated between a coil, partaking of the vibrations in the casing, and the seismographically mounted magnetic field. The measurement of velocity provides a better index of vibration energy than measurement of amplitude because energy is proportional to the square of velocity.

The interference-detector units and the amplifying system have a relatively flat frequency-response characteristic extending into the upper range of audibility. Thus a more faithful reproduction of the turbine noises is obtained than can be expected from long rods placed against the metal parts, with their attendant poor ear-coupling characteristics and natural period limitations.

Several detector units scattered about the turbine at strategic points can be used if desired. These, connected to a high-gain amplifier through a selective switching mechanism, will give a loud-speaker sound many times greater than the usual listening rod. The tremendous amplification available is of particular value when starting the turbine at which time little or nothing can be heard with a listening rod.

Mechanical Characteristics Recorded

Shaft Eccentricity. The eccentricity recorder measures the eccentricity of the turbine rotor shaft on the prolongation of the shaft extending beyond the bearing. Several temperature and loading factors will cause the shaft to assume a curvature for varying lengths of time. Some of these factors are (1) unequal application of heat around the periphery of the shaft, (2) sudden increments of heat (sudden loads) causing the shaft to distort by unequal absorption of heat, (3) rubbing of the shaft, and (4) misalignment of the bearings.

Such a curvature, occurring throughout the length of the rotating shaft between the bearing supports, not only produces undue mechanical stresses in the shaft, but will reduce the clearances between the shaft and the packing glands, and between the wheel rims and the diaphragm nozzles. Above a certain degree of

curvature, actual rubbing will take place, causing the packing glands to rub off, thus increasing the steam leakage and reducing the efficiency of the machine, and in extreme cases, causing material damage.

Due to this curvature, shown in Fig. 2, the end of the shaft projecting from the bearing will have a displacement of its center, and consequently of its periphery, once every revolution. Thus, the gap between a point A, stationary in space, and the surface of the shaft, will increase and decrease, once per revolution, by an amount dependent on the degree of curvature. It is at this point

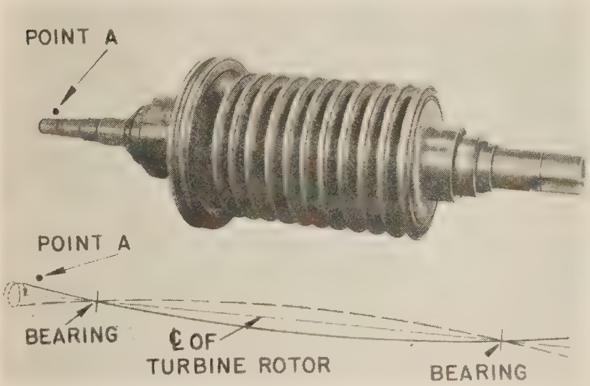

Fig. 2 Heat Distortion of Turbine Rotor

and by a measure of this varying gap that eccentricity is determined.

Bearing Vibration. The vibration-amplitude recorder shows the vibration amplitude of each of several bearings on the turbine-generator set in repetitive sequence. The vibration present in any particular bearing will vary from time to time. Some of the factors affecting this variation are (1) alignment of the bearings, (2) unbalance of rotating parts, (3) distortion of shaft during starting, (4) load, and (5) structural rigidity.

Quite apart from the damage to the bearing caused by excessive vibration, it has long been recognized that an abnormal increase in bearing vibration is an indication of changed conditions, generally leading to trouble.

Shell Expansion. The expansion recorder shows the change in the length of the turbine shell or casing, when more or less heat, in the form of steam, is admitted to the turbine. Changes in turbine load, and especially the enormous change from room temperature to operating temperature when starting the turbine, produce these variations in the length of the shell.

In order to retain the small clearances between the rotating and stationary parts of the turbine, it is necessary to have the shell expand axially in unison with the shaft. Should the expanding shell stick in the foundation slideways, terrific stresses would be set up and rubbing between the stationary and rotating parts might occur. It is of particular interest to know, in addition to the amount of expansion, whether the expansion is taking place uniformly or in a jerky manner.

Typical Starting Sequences

Fig. 3 shows the simultaneous eccentricity, vibration amplitude, and expansion records of a typical starting sequence taken on the high-pressure unit of a 160,000-kw turbine-generator set before the turbine bearing misalignment, discussed later, was corrected. Reading the records in the direction of time progress, from right to left, there are three distinct operation phases.

First is the period of turning-gear operation, when the turbine is being turned over very slowly, about $1\frac{1}{3}$ rpm by a motor drive. During this phase of operation, the width of the band record, irrespective of its vertical position on the chart, is a measure of the amount of eccentricity. This is shown at the upper right-hand corner in Fig. 3. In this range, one small division on the chart represents 0.001 in. Not until the turbine is rolled by steam, at the completion of the turning-gear phase, does the recorder switch to the $^{14}/_{1000}$-in. scale shown at the extreme left of the record. During turning-gear operation, this record shows the relatively small eccentricity to vary from 0.005 to 0.007 in. During this same interval, the vibration-amplitude recorder

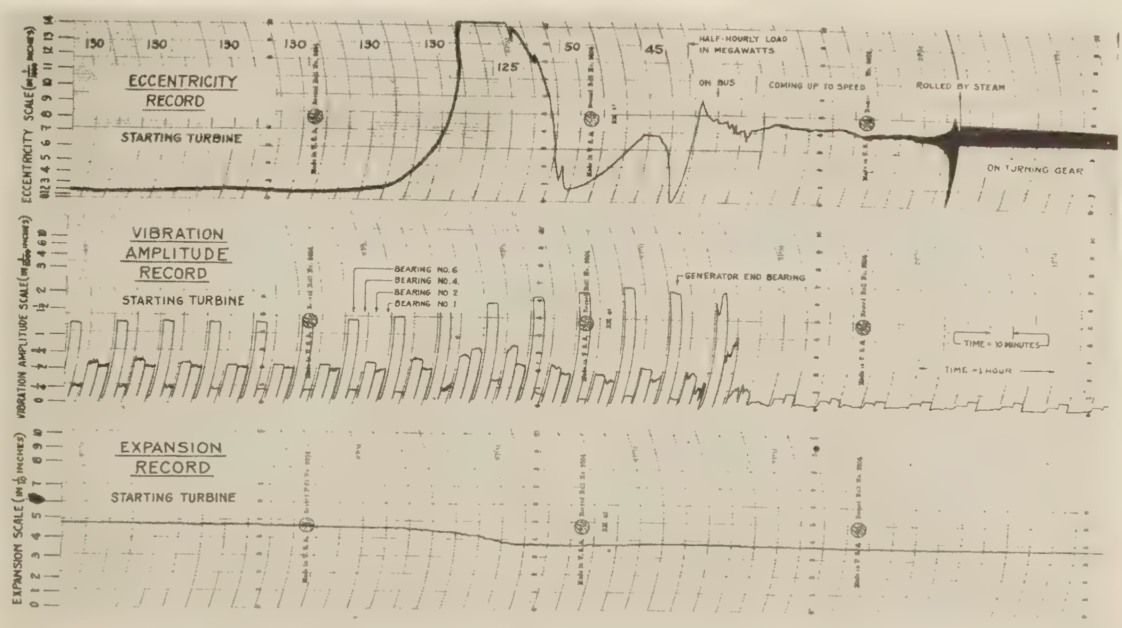

Fig. 3 Records of a Starting Sequence on a 160,000-Kw Steam-Turbine Generator

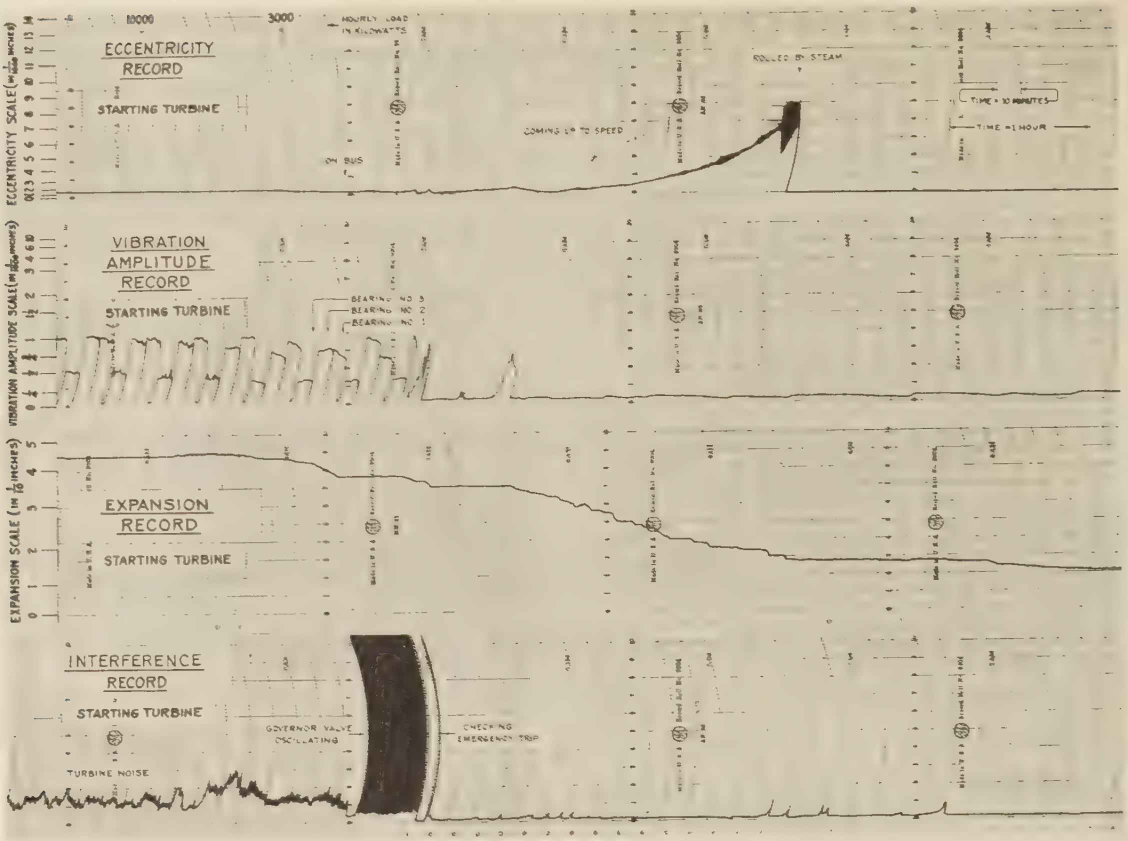

Fig. 4 Records of a Starting Sequence on a 20,000-Kw Mercury-Vapor-Turbine Generator

shows less than 0.00025 in. vibration amplitude. This vibration is structure-borne from adjoining machines. The expansion remains constant during this turning-gear period. Since it takes many days for a turbine of this size to cool to room temperature, an expansion of approximately 0.37 in. remains, even though the steam has been shut off for 24 hr.

The second operation phase, that of bringing the turbine up to speed, lasts from the time when the turbine is first rolled by steam until it is phased-in and the generator closed on the bus. During this period, the record shows the eccentricity to have increased but slightly, from 0.007 to 0.008 in. Several sharp variations in eccentricity occur prior to putting the machine on the bus. These mark the passage of the machine through a structural vibration resonance, and the effect of this resonance is clearly shown on the vibration-amplitude record in Fig. 3 immediately beneath the eccentricity record. At synchronous speed, this resonance has disappeared. The vibration amplitude does not become appreciable until the turbine has attained a speed of approximately 1500 rpm.

The final operation phase of a starting sequence is the assumption of load by the turbine. The most interesting phenomena occurred during this period, especially in connection with eccentricity. On this particular 160,000-kw turbine, the assumption of load almost invariably caused the shaft eccentricity to first decrease to zero and then increase to a value roughly proportional to the rate of load increase. This sudden decrease to zero eccentricity is probably due to a reversal of shaft curvature.

When the load is held relatively constant, the curvature works out of the shaft, as the forces and temperatures attain relative constancy. As shown in Fig. 3, this reduction is abruptly interrupted by the rapid assumption of much greater loads. When the load again becomes constant, the curvature works out of the shaft almost as fast as it went in, and the eccentricity falls to and remains at a relatively low value.

One of the most startling observations appearing on these and similar records is that the bearing vibration amplitude does not respond, except in a very minor degree, to a large increase in eccentricity. It appears as though the rotor acted like a free floating body with little or no reaction on its points of support.

The records obtained during the starting of a small turbine are somewhat different from those just discussed for a large turbine. Fig. 4 shows a typical starting sequence on a 20,000-kw Emmet mercury-vapor turbine. There is no turning-gear mechanism on this turbine. The eccentricity present when first rolled by mercury vapor disappears almost entirely while coming up to speed, nor does it show any appreciable increase upon the subsequent assumption of load. The turbine supervisory instruments on the mercury-vapor turbine have, in addition to the eccentricity, vibration-amplitude, and expansion recorders, an interference recorder which records the noises present in the turbine shell instead of making them audible by means of a loud-speaker system. The sounds produced by the release of the emergency oil-trip mechanism and the oscillation of a governor valve show up clearly.

Effect of Turning-Gear Mechanism

It has been known for some time that the rotor shaft of a large turbine will take on a semipermanent curvature when permitted to stand at rest for a few hours or more due to the difference in temperature caused by convection currents between the top and

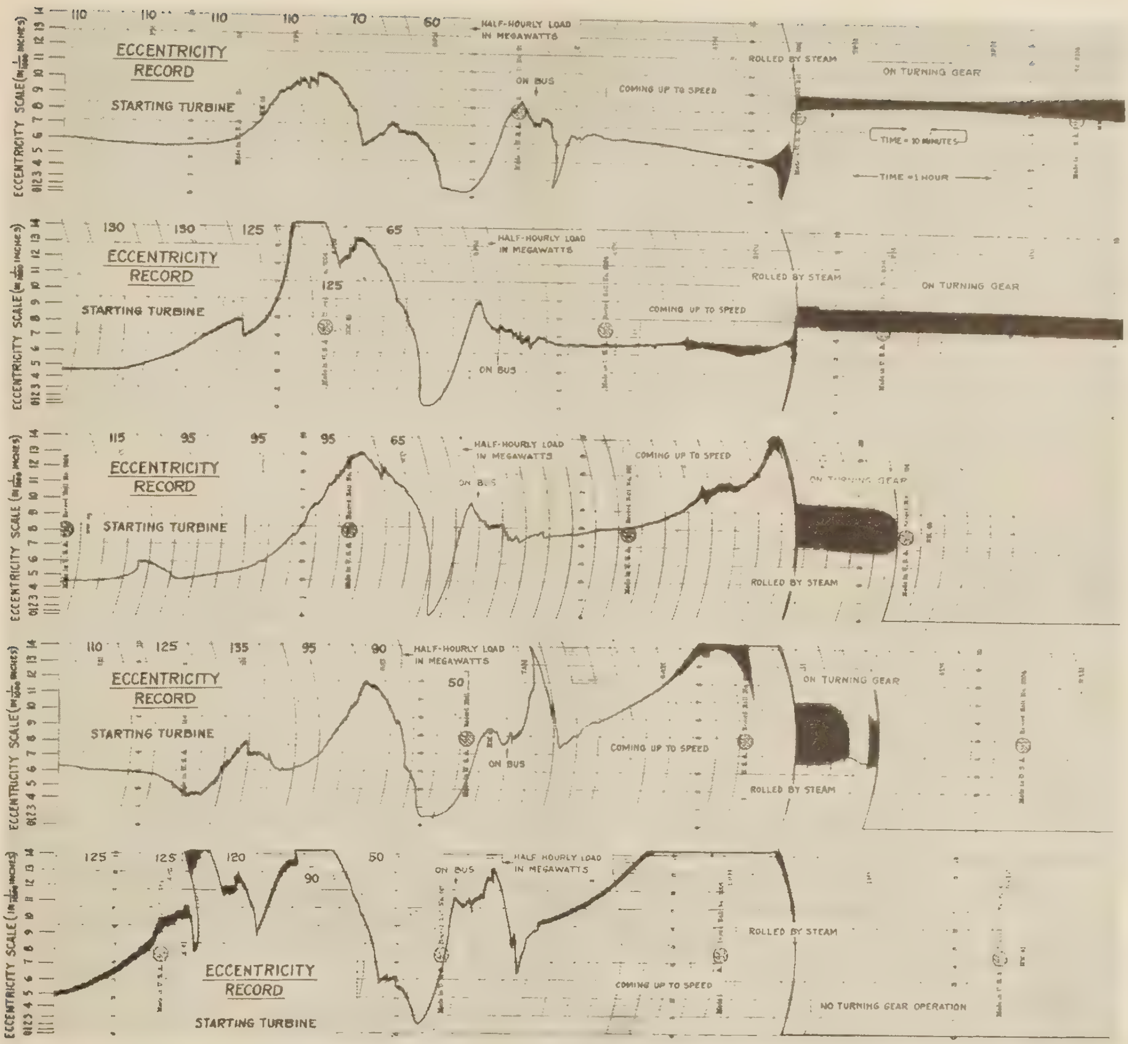

Fig. 5 Eccentricity Records of Several Starting Sequences on a 160,000-Kw Steam-Turbine Generator

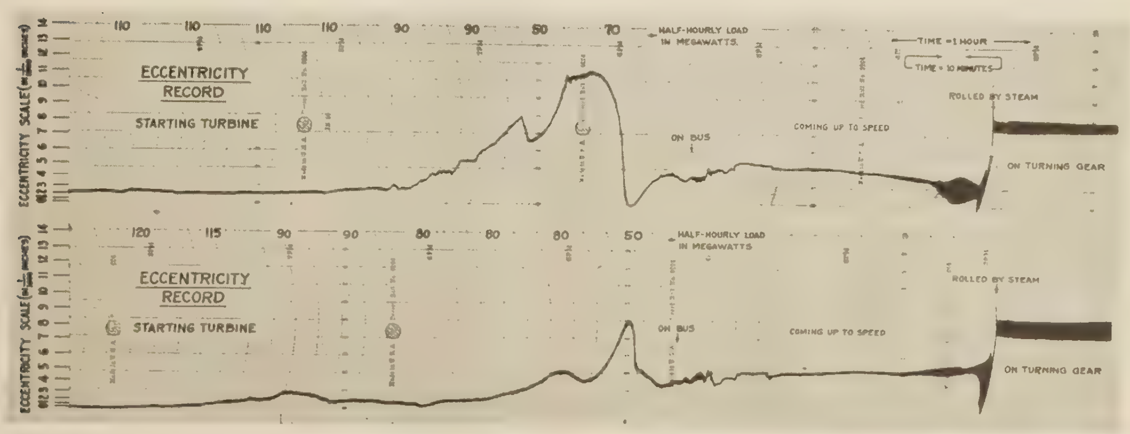

Fig. 6 Eccentricity Records of Starting Sequences Taken After the Correction of the Bearing Misalignment of a 160,000-Kw Steam-Turbine Generator

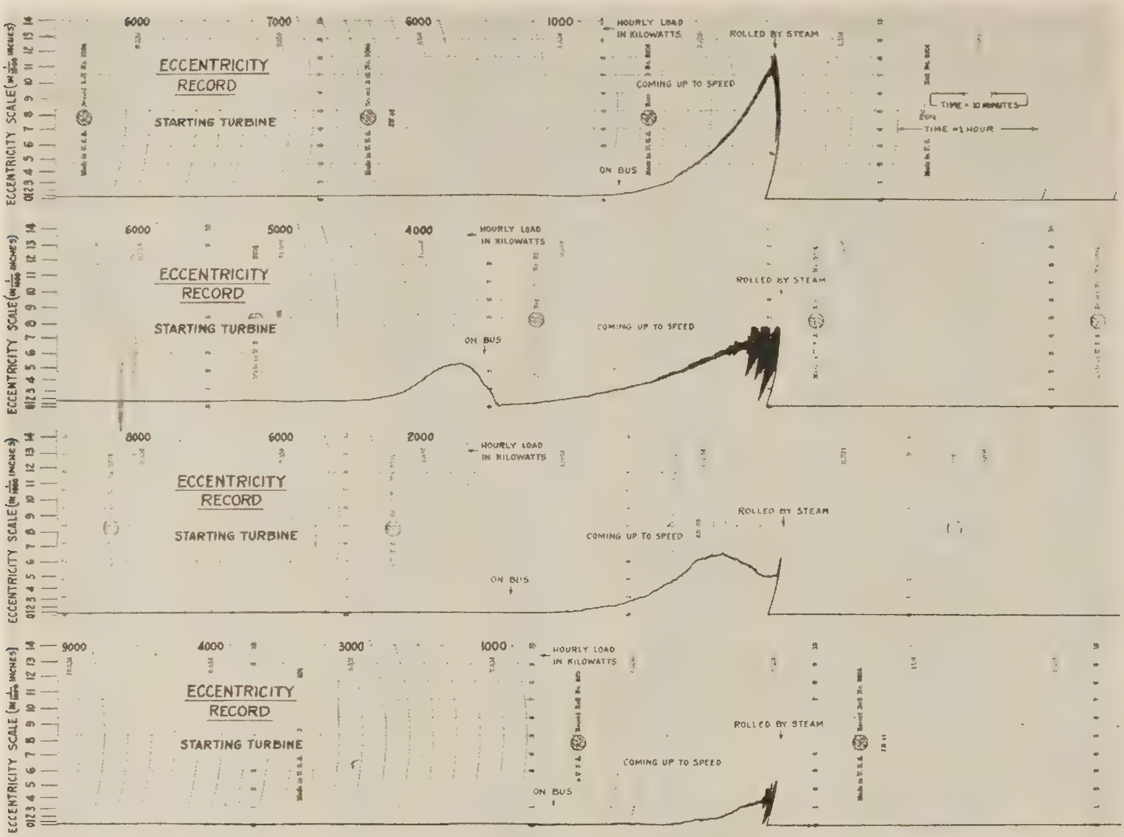

Fig. 7 Eccentricity Records of Several Starting Sequences on a 20,000-Kw Mercury-Vapor-Turbine Generator

bottom of the turbine casing. It is to eliminate or work out this assumed curvature that the aforementioned turning-gear mechanisms have been installed on many turbines. That the duration of turning-gear operation, and the amount of eccentricity present when the turbine is first rolled by steam, have a decided effect on the subsequent values of eccentricity is clearly shown by the records on Fig. 5. The longer the turning-gear period, the smaller the eccentricity at the end of this period, and in consequence the less the eccentricity during the coming-up-to-speed period. To a lesser degree, this diminished eccentricity level carries over into the assumption of load phase.

The eccentricity records shown in Fig. 5 were all made prior to the correction of the bearing misalignment of this particular machine, while those shown in Fig. 6 were made after this turbine was overhauled and the misalignment condition rectified.

These later records, Fig. 6, show the same general contour and response characteristics to a starting sequence as those appearing in Figs. 3 and 5. While the eccentricity values on these later records are less than before, it will be noticed that the rates of load assumption are also smaller. Nor did the vibration amplitude of the bearings give any markedly different response to an increase in eccentricity from formerly.

This comparison of several eccentricity starting records shows that approximately the same time interval elapsed between the start of the turbine by steam and its connection to the bus, regardless of the previous length of turning-gear operation or the amount of eccentricity. These starts were made in accordance with the standard practice recommended by the manufacturer. Providing other factors such as vibration and expansion are satisfactory, it is quite possible that a turbine can be put on the line in less than the usual time interval if the eccentricity is less than a predetermined value. When, however, the turning gear mechanism is not used at all, the record for which is shown at the bottom of Fig. 5, more than the usual amount of time should be taken.

Turbine Records Are Similar

On any given turbine, similar starting sequences or loading cycles give similar records. The general contour of the eccentricity records which are shown in Figs. 5, 6, and 7 are very similar. While the eccentricity response and values on the smaller turbine, Fig. 7, are markedly different from those on the large turbine, it is a common characteristic of both that with continued operation at uniform heat flow the eccentricities quickly diminish.

Effect of Sudden Loading

During long periods of operation at relatively constant load the eccentricity remains uniform and of small amplitude. Prior to the correction of the bearing misalignment, any sudden increase of load produced a rapid and distinct change in eccentricity. The top record in Fig. 8 shows the effect of jumping the load from 50,000 kw to 90,000 kw in less than 10 minutes. A very rapid increase in eccentricity was produced by this action and an almost equally rapid decrease occurred when the load steadied at the higher value.

After the turbine was overhauled and the bearing alignment corrected similar sudden increases in load, from 50,000 to 90,000

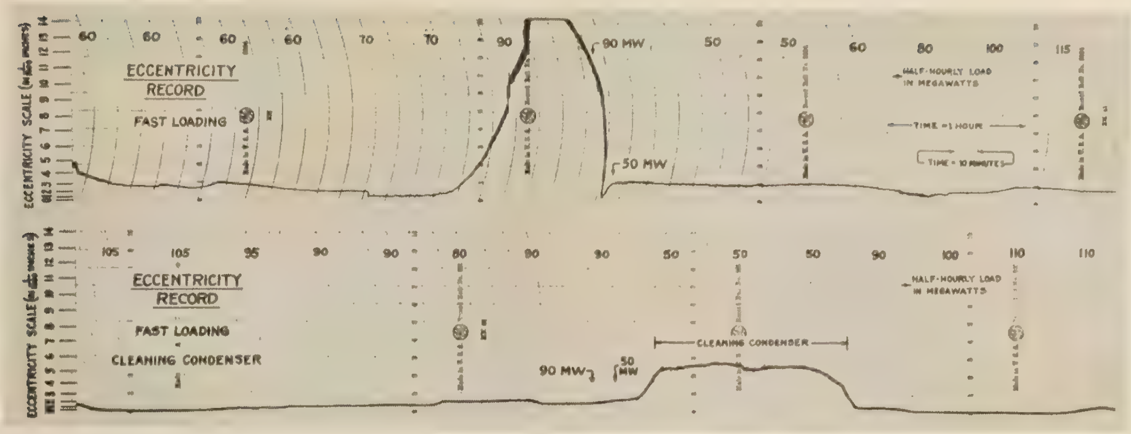

FIG. 8 ECCENTRICITY RECORDS OF SUDDEN LOADING AND EFFECT OF CONDENSER CLEANING DURING OPERATION OF A 160,000-KW STEAM-TURBINE GENERATOR

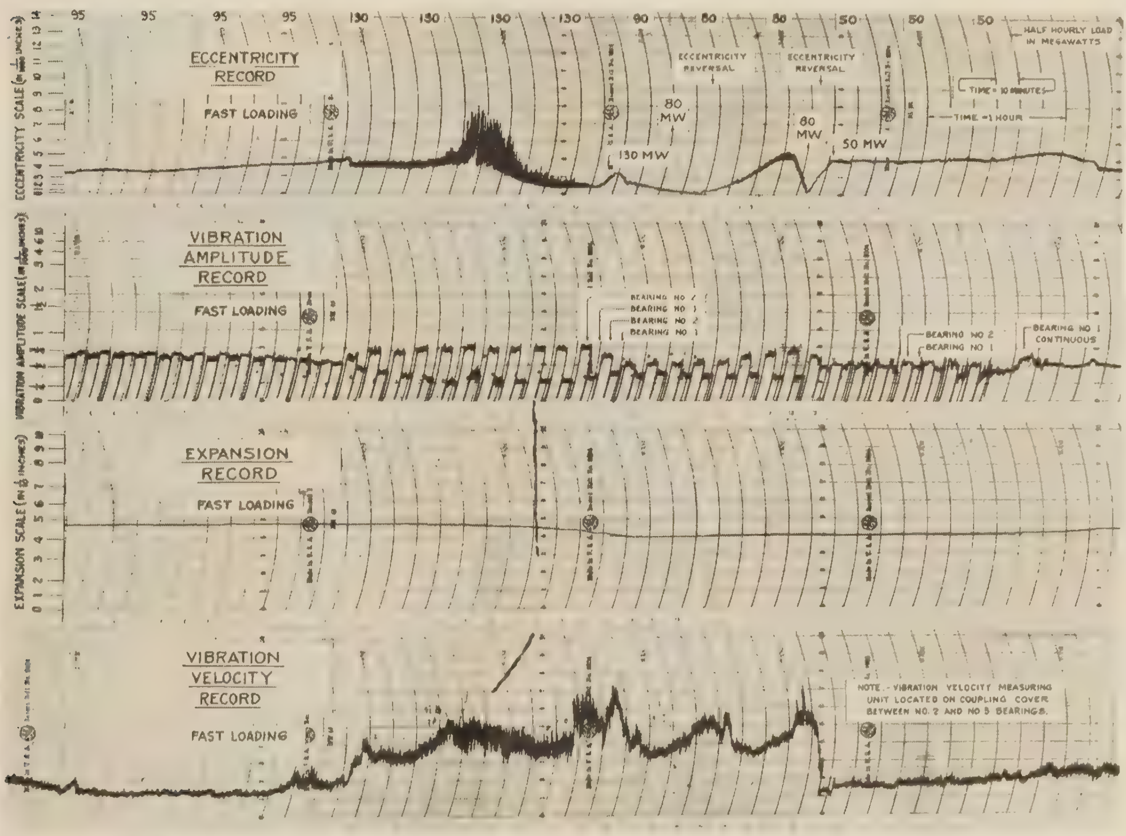

FIG. 9 RECORDS OF SPECIAL SUDDEN-LOADING SEQUENCE ON A 160,000-KW STEAM-TURBINE GENERATOR

w, produced no eccentricity response at all. This is shown in the lower record of Fig. 8.

EFFECT OF VACUUM VARIATIONS

This same lower record in Fig. 8 shows a very interesting eccentricity response when the load was reduced to approximately 50,000 kw to permit condenser cleaning operations. The condenser on this machine is so designed that water can be removed from successive sections for the purpose of cleaning the tubes while the turbine is operating. This action causes changes in the loading of the turbine-exhaust flange as well as variation of the temperature in the exhaust. The readjustment within the turbine to meet these changed conditions is shown on the lower record of Fig. 8.

EFFECT OF BEARING MISALIGNMENT

Subsequent to certain loading sequences and at definite loading values, an abnormal type of eccentricity record was obtained on

the 160,000-kw turbine. This condition appeared on the record chart shown in Fig. 9 as a wide, irregular band quite at variance with the normal relatively narrow line. There was nothing in the detecting or recording circuits which could of themselves cause this type of record. Investigation showed the phenomenon to be caused by the shaft itself. Oscillograph records disclosed that the end of the high-pressure shaft projecting beyond No. 1 bearing, where the eccentricity is measured, had, in addition to the cyclical eccentricity, quite irregular and random motions which were taking place at frequencies below and above the 1800-rpm, 30-cycle rotation frequency.

A special loading sequence was applied to the turbine and the records shown in Fig. 9 were obtained. The first sudden load application from 50,000 to 80,000 kw produced the normal change in eccentricity. However, the change took place in the form of a reversal of the curvature so that on the record it appears as a decrease to zero followed by an increase. When the load reached 80,000 kw and remained stationary at that point, the eccentricity again reduced to zero and the direction of curvature may have reversed once more. Prior to the beginning of this special loading sequence, the vibration-amplitude time switch had been reconnected so that the vibration of bearings Nos. 1 and 2, at the front and rear ends of the high-pressure unit, would appear in repetitive sequence rather than the usual 1-2-4-6-bearing sequence. In addition to this change, a vibration-velocity measuring instrument was attached to the steel cover of the jaw-type semirigid coupling between the high-pressure and low-pressure units, that is, between bearings Nos. 2 and 3. This instrument was attached to a recorder so that a record of the vibration velocity in the coupling cover plate could be obtained simultaneously with the usual records.

As soon as the load and eccentricity started to change, the vibration amplitude of No. 1 bearing decreased, while that of No. 2 bearing increased. In addition, the vibration velocity in the coupling cover plate showed a marked increase. After the load had steadied at 80,000 kw, the vibration amplitudes of Nos. 1 and 2 bearings again approached equality. During the second phase of the special loading sequence, the load was increased from 80,000 kw to 130,000 kw and then held constant. Shortly after reaching the 130,000-kw level, the wide-band record previously mentioned began to appear. After a little more than 1 hr at this load level, the wide band diminished appreciably. In the same manner as before, the vibration amplitude of No. 1 bearing decreased and No. 2 increased. By the time the load was decreased to 95,000 kw, almost 2 hr after reaching the 130,000-kw level, Nos. 1 and 2 bearings again had approximately the same vibration amplitude and the vibration velocity on the coupling cover plate had decreased to its initial value.

The decrease in vibration amplitude of bearing No. 1, when an increase was expected, coupled with the random motion of the end of the high-pressure shaft led to the conclusion that, during these load-level increases, the high-pressure shaft was partly lifted clear of No. 1 bearing, and that the whole high-pressure unit was operating as an outboard shaft from No. 2 bearing. This action would take place if No. 3 bearing were out of alignment and the jaw-type semirigid coupling between the low-pressure and the high-pressure units was unable to correct the misalignment between the two shaft sections when subject to increased loads. The jaw coupling became, in effect, a very rigid coupling and the greater weight of the low-pressure unit acting through this rigid coupling and utilizing No. 2 bearing as a fulcrum, lifted the outboard end of the lighter high-pressure unit off No. 1 bearing. The vibration-velocity record showed greatly increased stress in the coupling. This reasoning would require No. 3 bearing to be low and it was predicted that such was the case. When the turbine was subsequently taken down for overhaul, it was found that No. 3 bearing had settled 0.021 in.

Conclusions

1 Shaft eccentricity provides a quicker and much more definite measure of the mechanical conditions within a turbine than the former criterion of bearing vibration.

2 The reduction in shaft eccentricity due to turning-gear mechanisms, may, when indicated by turbine supervisory instruments, be utilized as one of the factors to permit quicker starting.

3 Records of shaft eccentricity, bearing vibration, and shell expansion made with turbine supervisory instruments, supply sufficient data for the correct diagnoses of complicated troubles such as the change in bearing alignment mentioned in this paper.

4 The permanent records supplied by these instruments not only provide the operator with immediate information on the mechanical conditions within the turbine, but permit the performance to be studied and analyzed later, at leisure.

5 While the instruments were developed primarily for remote outdoor operation of turbines, these records show that they are of prime value on any turbine.

Superposed-Turbine Regulation Problem

By A. F. Schwendner[1] and A. A. Luoma,[2] Philadelphia, Pa.

This paper gives a description of the hydraulic governing system used on Westinghouse superposed turbines. It also attacks from a theoretical standpoint some of the governing problems encountered on superposed turbines. The problems specifically attacked are (1) stable synchronizing, and (2) stable operation on the line with an exhaust-pressure regulator in action.

Introduction

The superposed back-pressure turbine presents problems of control which are more difficult than those encountered so far for condensing turbines. The heat drops are usually low, leading to enormous steam flows even for moderate capacities. Since there is no limitation imposed in connection with exhaust area, 3600-rpm turbines can be used up to the limit of the generator capacity. This leads to an abnormally small spindle inertia when compared to condensing turbines.

Regardless of the great difference in turbine characteristics, the regulation requirements are identical with those of condensing turbines. The regulating requirements are: (1) The governor should be stable when the turbine is being synchronized, and (2) the governor must be stable when the turbine is running on the line and carrying load.

With the turbine on the line and carrying load, changes in steam flow will depend on exhaust, steam demand, and the type of regulation required by the overall efficiency of the particular power plant. The superposed turbine may run as an individual unit exhausting steam into the low-pressure boiler header. The steam flow, and consequently load carried by the superposed turbine, is determined by the power-plant load and steam demand, and is controlled by the governor speed changer. In another case, the superposed turbine only supplies steam to one or several low-pressure turbines. In this case the governor will have to respond to the varying flow demand of the low-pressure units and keep the pressure in the low-pressure header within close limits. The load on the superposed turbine will be determined by the steam demand of the low-pressure units, and will be controlled by a speed governor which will also respond to the exhaust-pressure changes. Finally, we have the combination where the superposed turbine is electrically tied to one or more low-pressure turbines. In this case the governing valves of the low-pressure turbine are wide open and the combination acts as one unit with the superposed-turbine governor controlling load and speed set by the governor speed changer. Through the electrical tie, the inertia of the low-pressure-turbine spindle is more or less added to the superposed-turbine spindle. Thus, the combination will have the governing characteristics of a condensing unit, and for this reason it will be considered beyond the scope of this paper.

Regulation requirements specify stability, which means that there should be no periodic speed or load change due to the governor itself. Stability can be obtained by proper relation between the governor-speed variation, its rapidity of response, inertia of the turbine spindle and generator rotor, governing-valve sizes, and other factors. The governor and connecting linkage must also be free to a considerable degree from friction and back lash.

To be able to obtain the proper relation between the turbine and control parts, a fundamental analysis of the problem must be made, while careful attention to design details will greatly reduce friction and back lash.

The remainder of this paper is divided into two general divisions. The first is devoted to a description of the hydraulic-governing system used on Westinghouse superposed turbines. The second part is devoted to a theoretical analysis, first, of synchronizing stability, and, second, of stability on the line with an exhaust-pressure regulator in action. Theoretical criteria of stability are derived for each of these cases. While the governing system analyzed is that used on Westinghouse turbines, the method used is applicable to other governing systems. Some of the conclusions also apply to other governing systems.

The problem of stable operation on the line without an exhaust-pressure regulator in action presents no special difficulties and will be disregarded.

Description

In the hydraulic governing system, instead of transforming speed into motion through the medium of a flyball governor, the shaft speed is measured by the pressure in a chamber surrounding the shaft. This pressure is maintained by the centrifugal force of the oil in the inclined hole drilled in the turbine shaft connecting the pressure chamber to drain, as shown in Fig. 1. Oil is supplied through an orifice into the pressure chamber from the main oil pump located on the turbine shaft next to the inclined hole, otherwise called the governor impeller. The small amount of oil flowing through the orifice into the pressure chamber maintains a pressure which varies as the square of the turbine speed. Excess oil supplied through the orifice above that required to maintain the pressure in this chamber passes beyond the center of the shaft and then out to drain.

To increase the sensitivity and response of the servomotor to very small speed changes, the originally small pressure change of the governor impeller is magnified in the governor transformer. The transformer consists of a bellows which is opposed by the speed-changer spring and the pressure in the small control chamber on top of the transformer relay. The pressure in this

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control chamber depends on the pressure change of the governor impeller and the difference in the areas of the bellows and transformer relay. The initial oil pressure is carried by the speed-changer spring. The control chamber is connected through drilled holes to the regulating annulus located at the middle of the transformer relay. This, in turn, may communicate with high-pressure oil above and with the drain connection below, depending on the motion of the transformer relay. Assuming the effective area of the bellows to be 10 sq in., and the differential area of the transformer relay 1 sq in., an increase of 1 lb per sq in. of oil pressure below the bellows will produce an increase in the regulating pressure of 10 lb per sq in. The transformer relay is kept revolving at all times by a jet of oil directed against a turbine element mounted on the transformer relay. This is done to reduce friction to a minimum and to make the mechanism highly sensitive. The regulating pressure is admitted to the bellows chamber of the governor servomotor. The regulating pressure acts on the effective area of the bellows in a downward direction and is opposed by a tension spring between the bellows and return link. The main relay is moved downward by the oil pressure above the relay, which is supplied through an orifice in the relay bushing. The compression spring below the main relay moves the relay in an upward direction. The movements of the main relay are controlled by the pilot relay attached to the bellows. An increase in the regulating pressure moves the bellows in a downward direction. The oil pressure above the main relay will increase due to the decreased flow between the main and pilot relay. The increased oil pressure will move the relay downward. The main relay in this position admits high-pressure oil under the operating piston and allows the oil on top of the piston to discharge.

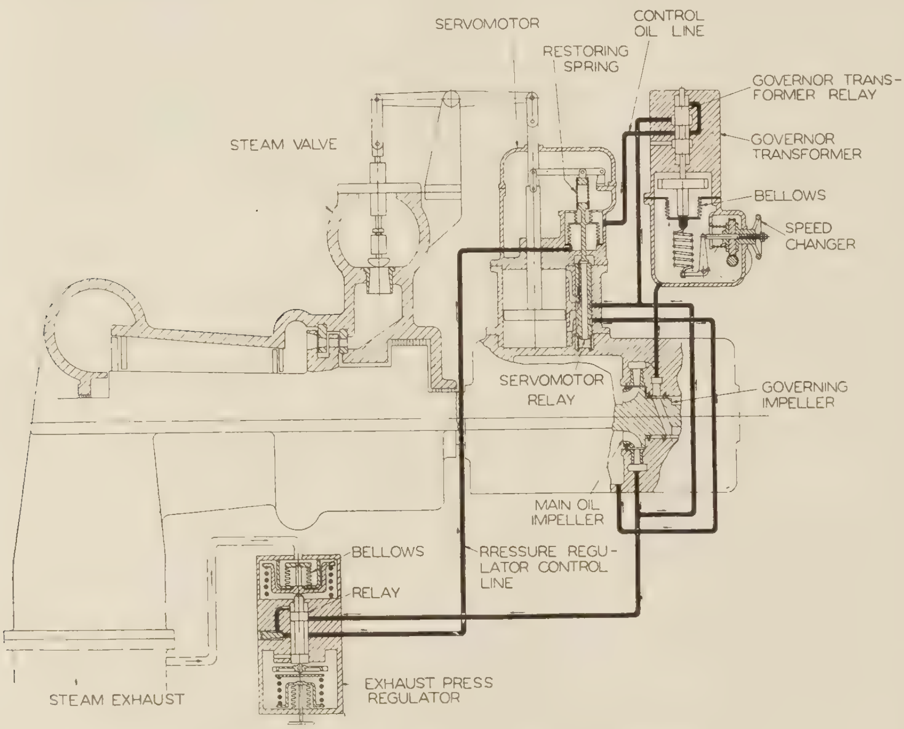

Fig. 1 Hydraulic Governing System for Turbines

The resultant upward motion of the operating piston is checked only when the return link has increased the spring tension by the same amount that the regulating pressure forced the bellows downward. This will bring the bellows, pilot and main relays into neutral position. Tracing through a change in speed, the system operates as follows: An increase in speed produced an increase in governor-impeller pressure which, in turn, increased the regulating pressure. The increased regulating pressure moves the operating piston upward which, in turn, closes the steam-admission valves.

To make the governor respond to an impulse other than speed, control pressure regulated by an exhaust-pressure regulator can be admitted below the bellows of the governor servomotor. A decrease in control pressure underneath the bellows will have the same effect as an increase in the regulating pressure above the bellows. As an increase in the exhaust pressure requires closing of the governing valves, the exhaust-pressure regulator has to be so constructed that an increase in the exhaust pressure will produce a decreased control pressure. The exhaust pressure regulator, as shown in Fig. 1, has a bellows chamber which is connected to the turbine exhaust. The control chamber of the regulator relay is located on the top and the control pressure acts on the regulator relay in the same direction as the exhaust pressure. With the regulator relay in its neutral position, the force of the exhaust pressure on the bellows, plus the force of the oil pressure on the regulator relay, must balance the spring forces at the bellows and below the regulator relay. Since the spring force is constant at the regulator relay neutral point, an increase in exhaust pressure will decrease the control pressure. The effective areas of the bellows and regulator relay are so selected that the required exhaust-pressure change will create a control-pressure change which can move the operating piston from full-open to closed valve positions. The regulator relay is also kept revolving in the same manner as the transformer relay.

To be able to follow the smallest speed or pressure changes, the hydraulic system is so constructed that pressure changes can be transmitted with very little actual oil flow. The relays which have to follow small pressure changes are kept revolving to reduce friction. The parts which have to respond to larger or transformed pressure changes have bellows with comparatively large effective areas and move a pilot relay to reduce the friction and flow reaction of the large main relay. Oil passages in the governor are made large enough to supply large amounts of oil with small pressure drops.

Synchronizing Stability Problem

The problem discussed in this paper involves the finding of the fundamental relationship between certain constants of the governor, governor servomotor and turbine, that must be satisfied to have stable governing off the line at no load. A superposed turbine generator when being synchronized has its back-pressure regulator, if provided, out of action, and hence the

problem of its synchronization in its general aspects is identical with that of a straight condensing turbine generator.

The symbols and terms used are listed in the nomenclature. The governing system to be analyzed is shown in Fig. 1. The back-pressure regulator is assumed to be irresponsive to back-pressure variations. Let us assume that the turbine generator is running at exactly constant speed and then a resisting torque ΔM is applied instantaneously to the rotating parts. Considering the system linear, then in general a speed disturbance of the type

$$\omega = A_0 + A_1 e^{\alpha_1 t} \cos (q_1 t + \epsilon_1) + A_2 e^{\alpha_2 t} \cos (q_2 t + \epsilon_2) + \ldots.$$

will be produced. The term A_0 represents the permanent speed change due to the resisting torque ΔM, and the remaining terms represent the accompanying transient disturbance. If the transient dampens out, the system is stable. In order that the transient dampen out, α_1, α_2, α_3, etc., must each be negative and, therefore, the problem in its simplest aspects reduces to predicting α_1, α_2, α_3, etc. The differential Equation [7a] derived in Appendix 1, represents the behavior of the governing system in Fig. 1, when the turbine generator is off the line and an instantaneous resisting torque ΔM is applied to the spindle, the back-pressure regulator being out of action. This equation is

$$\frac{d^4\omega}{dt^4} + \left(\frac{1}{T_x} + \frac{1}{T_I}\right)\frac{d^3\omega}{dt^3} + \left(\frac{1}{T_x T_I} + \frac{1}{T_x T_y}\right)\frac{d^2\omega}{dt^2} + \left(\frac{1}{T_x}\frac{1}{T_y}\frac{1}{T_I}\right)\frac{d\omega}{dt} + \frac{T_M}{T_x T_y T_I}\omega = \frac{-T_M}{T_x T_y T_I}\frac{\Delta M}{M} \ldots\ldots\ldots [7a]$$

The significance of the various terms is as follows: T_x is the time constant of the governor transformer and is a measure of the rapidity of response of the governor transformer. The smaller the value of T_x, the faster will be the action of the governor transformer. Likewise, T_y is the time constant of the servomotor. The time constant T_I is due to the steam entrapped between the first governing valve and nozzle inlet, and is a measure of the time lag due to this entrapped steam. The value of T_M is a measure of the correcting effect of the governing system due to speed changes and is the acceleration in radians per sec per sec produced by the torque developed when the turbine speed drops 1 radian per sec, this latter torque being equal to M. The solution of Equation [7a] determines the system stability. A convenient method for checking the stability without actually solving the differential equation is as follows: Differential Equation [7a], disregarding the term on the right side, is of the form

$$\frac{d^4\omega}{dt^4} + C_3\frac{d^3\omega}{dt^3} + C_2\frac{d^2\omega}{dt^2} + C_1\frac{d\omega}{dt} + C_0\omega = 0$$

where the C terms are constants.

This equation represents a stable system if all the C terms are positive, and, if $C_1C_2C_3 > C_1^2 + C_0C_3^2$. The first condition is satisfied. The second condition gives

$$\left(\frac{1}{T_I + T_x}\right) + \frac{1}{T_y\left(1 + \frac{T_x}{T_I}\right)^2} > T_M \ldots\ldots\ldots [9a]$$

From criterion [9a] it is evident that decreasing T_M, which means increasing the inertia of the rotating parts of the turbine generator, or increasing the regulation of the governor, or decreasing the size of the governing valve that is in operation when synchronizing occurs, improves stability. Also from criterion [9a] it is apparent that decreasing T_x and T_y, or in other words making the governor transformer and governor servomotor faster acting, also has a beneficial influence on the stability of the system. If T_x is small, as is usually the case, decreasing T_I, or in brief decreasing the volume of entrapped steam between the governing valve and nozzle inlet, has a beneficial effect on stability. If, however, T_x is large, then decreasing T_I may have a detrimental effect on stability.

In superposed turbines, T_M is usually quite large, and this necessitates careful design of the governor transformer and governor servomotor to obtain values of T_x, T_y, and T_I which are as small as possible. It is especially important to obtain small values of T_x and T_I.

Stability of Back-Pressure Governing. In analyzing the stability of the back-pressure control, the superposed turbine will be assumed to be running on the line at exactly constant speed. Under this assumption, the steam flow through the superposed turbine will be entirely under the control of the back-pressure regulator. In Appendix 2, the differential equation of the system is derived. This Equation [7b] is

$$\frac{d^3P_{sB}}{dt^3} + \left(\frac{1}{T_{xB}} + \frac{1}{T_{IB}}\right)\frac{d^2P_{sB}}{dt^2} + \left(\frac{1}{T_{xB}}\frac{1}{T_{IB}} + \frac{1}{T_{xB}T_y}\right)\frac{dP_{sB}}{dt} + \left(\frac{1}{T_{xB}}\frac{1}{T_y}\frac{1}{T_{IB}} + \frac{T_{MB}}{T_{xB}T_y}\right)P_{sB} = 0 \ldots\ldots\ldots [7b]$$

where T_{xB} is the time constant of the back-pressure regulator, T_y is the time constant of the governor servomotor, T_{IB} is the time constant due to the capacity effect of the superposed turbine exhaust-system volume, T_{MB} is the response constant of the back-pressure control system, and P_{sB} represents the variation in exhaust pressure from the initial value. The solution of Equation [7b] is of the form

$$P_{sB} = A_0 e^{\alpha_0 t} + A_1 e^{\alpha_1 t}\cos(q_1 t + \epsilon_1)$$

If α_0 and α_1 are each negative the system is theoretically stable. A convenient method of checking the stability of the system directly from the differential equation is as follows: The differential equation is of the form

$$\frac{d^3P_{sB}}{dt^3} + C_2\frac{d^2P_{sB}}{dt^2} + C_1\frac{dP_{sB}}{dt} + C_0P_{sB} = 0$$

which represents a stable system if the coefficients C are each positive and if $C_1C_2 > C_0$. The coefficients of Equation [7b] are all positive. The second criterion gives

$$\frac{T_y}{T_{xB}T_{IB}} + \frac{1}{T_{xB}} + \frac{T_y}{(T_{IB})^2} > T_{MB} \ldots\ldots\ldots\ldots [8b]$$

From criterion [8b] it is apparent that the following factors make the system more stable: Increasing the regulation of the back-pressure regulator; increasing the weight of entrapped steam per pound per square inch absolute, in the exhaust system, which for a given installation means a larger exhaust-system volume; decreasing the maximum value of φ_v by better governing-valve design. The preceding factors all decrease T_{MB}. The stability of the system can also be improved by decreasing the time constant of the back-pressure regulator T_{xB}, by increasing the servomotor time constant T_y, and by decreasing the exhaust-system time constant T_{IB}. The exhaust-system time constant decreases as the load on the low-pressure turbines is increased, because the factor φ_{tB} increases with load on the low-pressure turbines due to wider valve openings. On a given system, the only way of decreasing T_{IB} is to decrease the volume of the exhaust system, but this will increase T_{MB}, so that the net effect will be detrimental to stability. The assumption that the superposed turbine runs at exactly constant speed implies that

the superposed-turbine installation is tied together with a reasonably large system.

Nomenclature

a_{x1} = effective area of governor transformer bellows, sq in.
a_{x1B} = effective area of back-pressure-regulator bellows, sq in.
a_{x2} = area of end of governor transformer relay, sq in.
a_{x2B} = area of end of back-pressure-regulator relay, sq in.
a_{y1} = area of governor servomotor bellows, sq in.
a_{y2} = area of governor servomotor piston, sq in.
d_s = increase in density of steam entrapped between governing valve and nozzle inlet, due to increase in pressure P_s, lb per cu ft
d_{sB} = increase in density of entrapped steam in exhaust system due to increase in pressure P_{sB}, lb per cu ft
e = base of Napierian logarithm system
F_s = increase in flow through turbine due to increase in steam pressure P_s in steam entering nozzle, lb per sec
J = mass moment of inertia of turbine generator rotating parts, lb in sq sec
K_s = ratio between density and absolute pressure of steam entrapped between primary valve and primary nozzle inlet, lb per cu ft per lb per sq in.
K_B = ratio between density and absolute pressure of steam in exhaust system, lb per cu ft per lb per sq in.
K_x = total scale of governor-transformer spring and bellows, lb per in.
K_{xB} = total scale of back-pressure-regulator springs and bellows, lb per in.
K_{y1} = restoring spring scale of governor servomoter, lb per in.
K_y = total scale of governor-servomotor bellows and spring, lb per in.
M_s = increase in torque, in-lb per lb per sec increase in steam flow through turbine
M = torque produced by increase in steam flow through turbine due to decrease in turbine speed of 1 radian per sec, in-lb
ΔM = increment of resisting torque applied to turbine spindle, in-lb
P_0 = governing impeller pressure at speed ω_0, lb per sq in.
P_s = increase in pressure of steam entering primary nozzle above initial value, lb per sq in.
P_{sB} = increase in exhaust-steam pressure above initial value, lb per sq in.
P_{x2} = increase in governor-transformer regulating pressure above initial value, lb per sq in.
P_{x2B} = increase in back-pressure-regulator control pressure above initial value, lb per sq in.
r_y = ratio of restoring spring travel to piston travel on governor servomotor
t = time, sec
T'_x = governor-transformer time constant, sec
T'_{xB} = back-pressure-regulator time constant, sec
T'_y = governor-servomotor time constant, sec
T_I = time constant due to entrapped steam between primary valve and primary nozzle, sec
T_{IB} = time constant due to entrapped steam in exhaust system sec
T_M = governing-system acceleration constant 1/sec
T_{MB} = response constant of back-pressure control system, 1/sec
V_s = volume of entrapped steam between primary valve and primary nozzle inlet, cu ft
V_B = volume of back-pressure turbine exhaust system, cu ft
W_s = increase in weight of entrapped steam between primary valve and primary nozzle inlet due to increase in pressure P_s, lb

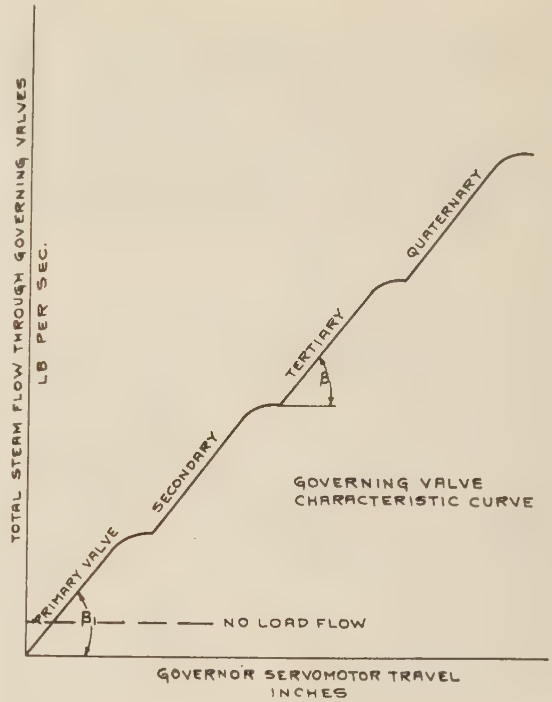

Fig. 2 Valve Characteristic Curve

W_{sB} = increase in weight of entrapped steam in exhaust system due to increase in pressure P_{sB}, lb
x = upward displacement of governor-transformer relay from neutral, in.
x_B = upward displacement of back-pressure-regulator relay from neutral, in.
y = upward displacement of governor servomotor relay from neutral, in.
z = downward displacement of governor servomotor piston from initial position, in.
φ_t = increase in flow through superposed turbine, lb per sec per lb per sq in. increase in pressure in nozzle inlet
φ_{tB} = increase in flow out of exhaust system, lb per sec per lb per sq in. increase in exhaust-system pressure
φ_x = flow constant of governor-transformer relay cu in. per sec per in. lift
φ_{xB} = flow constant of back-pressure-regulator relay cu in. per sec per in. lift
φ_v = slope of curve on Fig. 2 showing total steam flow through governing valves versus servomotor travel. Slope to be taken at the particular point considered.
φ_y = flow constant of governor-servomotor relay, cu in. per sec per in. lift
β_1 = angle shown in Fig. 2
β = angle shown in Fig. 2
ω_0 = initial turbine speed, radians per sec
ω = increase in turbine speed above initial speed, radians per sec

Appendix 1

Analysis of Governor Stability Under Synchronizing Conditions

In this appendix an analysis will be made of the governing system shown in Fig. 1, the turbine generator running off the line, and the back-pressure regulator being irresponsive to back-

pressure changes. The symbols listed in the nomenclature will be used.

Let the turbine generator be assumed to be running at exactly constant speed off the line at no load. If a small resisting torque ΔM be applied instantaneously to the spindle, a disturbance will be set up in the speed of the unit. The criterion of stability is whether this disturbance dampens out, or whether it continues to increase to a greater disturbance.

The method of analysis will be to set up the differential equations which apply to the component parts of the governor and then to solve these equations simultaneously to obtain the general differential equation of the governing system. From the general differential equation, either by solving it, or by mathematical criterions, we can determine whether the governing system is theoretically stable.

At any instant t after application of the resisting torque ΔM, let the speed of the unit be assumed ω radians per sec above the original speed ω_0. At this instant t, the governing-impeller pressure, assuming it to be proportional to the square of the speed, will have increased an amount $\frac{2P_0\omega}{\omega_0}$, the governor-transformer regulating pressure will have increased by an amount P_{x2}, and the governor-transformer relay will be above its neutral position by an amount x. Since the governor-transformer relay must always be in equilibrium under the various forces acting on it, we obtain, neglecting relay mass and damping, the following equation

$$\frac{2P_0a_{x1}\omega}{\omega_0} = a_{x2}P_{x2} + K_xx \qquad [1a]$$

The instantaneous rate of oil flow through the governor-transformer relay will be taken proportional to the departure of the relay from its neutral position and must be equal to the instantaneous rate of storage in the servomotor operating bellows, or

$$\varphi_x x = -a_{y1}\frac{dy}{dt} \qquad [2a]$$

Considering the equilibrium of the servomotor relay, and neglecting mass and damping we obtain

$$a_{y1}P_{x2} + K_yy + r_yK_{v1}z = 0 \qquad [3a]$$

The flow through the servomotor relay likewise will be taken proportional to its displacement from neutral position and is related to the servomotor piston velocity, thus

$$y\varphi_y = a_{y2}\frac{dz}{dt} \qquad [4a]$$

Next the flow of steam through the governing valves will be considered. The relationship of steam flow to servomotor travel is shown on Fig. 2. Taking $\varphi_{v1} = \tan\beta_1$, the increase in steam flow at instant t through the governing valve, above its initial value, will be $\varphi_{v1}z$. However, the instantaneous flow through the turbine will not be equal to the instantaneous flow through the governing valve on account of the volume of steam entrapped between the valve and nozzle inlet. If the total heat of the entrapped steam be assumed equal to that of the steam before the throttle, the increase in density of the entrapped steam will be proportional to the increase in absolute pressure P_s, or $d_s = K_sP_s$ and the increase in weight, and rate of increase of weight of entrapped steam will be given respectively by

$$W_s = V_sK_sP_s$$

and

$$\frac{dW_s}{dt} = V_sK_s\frac{dP_s}{dt}$$

The increase in flow through the turbine above its initial flow can be taken proportional to P_s or

$$F_s = \varphi_tP_s$$

Hence, for flow equilibrium we have

$$\varphi_{v1}z = V_sK_s\frac{dP_s}{dt} + \varphi_tP_s \qquad [5a]$$

Finally, the equilibrium of the rotating parts of the turbine-generator must be considered. Due to the increase in steam flow through the turbine, there will be a torque component of $M_sF_s = M_s\varphi_tP_s$. Because of the instantaneous acceleration $d\omega/dt$ there will be a component $J(d\omega/dt)$. Neglecting spindle damping, we obtain

$$M_s\varphi_tP_s = J\frac{d\omega}{dt} + \Delta M \qquad [6a]$$

Simultaneous differential Equations [1a] to [6a], inclusive, completely represent the behavior of the governing system during the transient disturbance. Solving these equations by standard methods we obtain the following single differential equation with ω as a function of t, or

$$\frac{d^4\omega}{dt^4} + \left(\frac{1}{T_x} + \frac{1}{T_I}\right)\frac{d^3\omega}{dt^3} + \left(\frac{1}{T_xT_I} + \frac{1}{T_xT_y}\right)\frac{d^2\omega}{dt^2} + \left(\frac{1}{T_xT_yT_I}\right)\frac{d\omega}{dt} + \frac{T_M\omega}{T_xT_yT_I} = -\frac{T_M}{T_xT_yT_I}\frac{\Delta M}{M} \qquad [7a]$$

where

$$T_x = \left(\frac{K_xa_{y1}^2}{\varphi_xa_{x2}K_y}\right)$$

is the time constant of the governor transformer,

$$T_y = \frac{K_ya_{y2}}{\varphi_yr_yK_{v1}}$$

is the servomotor time constant,

$$T_I = \frac{V_sK_s}{\varphi_t}$$

is the time constant of entrapped steam between governing valve and nozzle inlet, and

$$T_M = \frac{2P_0a_{x1}a_{y1}\varphi_{v1}M_s}{\omega_0a_{x2}r_yK_{v1}J}$$

is the turbine-generator acceleration constant. The turbine generator acceleration constant T_M is the acceleration produced by the torque developed by the increase in steam flow produced by decreasing the turbine speed 1 radian per sec, assuming a linear relation between the torque developed and the speed decrease. The torque

$$M = \frac{2P_0a_{x1}a_{y1}\varphi_{v1}M_s}{\omega_0a_{x2}r_yK_{y1}} = T_MJ$$

is that produced by decreasing the turbine speed 1 radian per sec. Solving Equation [7a] gives in general the solution

$$\omega = A_1e^{\alpha_1t}\cos(q_1t + \epsilon_1) + A_2e^{\alpha_2t}\cos(q_2t + \epsilon_2) - \frac{\Delta M}{M} \qquad [8a]$$

where A_1 and A_2 are constants determined by the initial conditions. The term $-(\Delta M/M)$ represents the permanent change in speed due to torque ΔM. The other two terms represent the transient disturbance in the speed. If α_1 and α_2 are both negative, the transient dampens out and the governing system is stable.

Appendix 2

Analysis of the Stability of the Back-Pressure Regulator

The system in Fig. 1 will be analyzed, and the terms and symbols listed in the nomenclature will be used. The turbine-generator will be assumed to be on the line running at exactly constant speed. Under this condition the governor of the unit will be solely under the control of the back-pressure regulator.

The same general method that was used in Appendix 1 will be used. Let us assume that the governing system is running under steady conditions, and the back pressure is holding exactly constant. Then, if some disturbance occurs, a transient will be set up which will be completely represented by the following simultaneous differential equations. Considering the equilibrium of the back-pressure regulator relay, and neglecting mass and damping of the relay, we obtain

$$P_{sB}a_{x1B} + P_{x2B}a_{x2B} + K_{xB}x_B = 0 \quad \text{[1b]}$$

The instantaneous rate of flow of oil through the back-pressure regulator relay will be taken proportional to the relay displacement from neutral, and must equal the instantaneous rate of oil storage in the governor-servomotor bellows. Hence

$$\varphi_{xB}x_B = a_{y1}\frac{dy}{dt} \quad \text{[2b]}$$

Considering the equilibrium of the servomotor relay and neglecting mass and damping of the relay, we obtain

$$P_{x2B}a_{y1} = K_y y + r_y K_{y1} z \quad \text{[3b]}$$

As represented in Appendix 1, the servomotor-piston speed is related to the servomotor-relay displacement. Thus

$$\varphi_y y = a_{y2}\frac{dz}{dt} \quad \text{[4b]}$$

The flow change through the governing valve for small lift variations can be taken proportional to the lift variation. The effect of entrapped steam between the governing valve and nozzle inlet will be neglected. The density of the steam in the exhaust system of the back-pressure turbine can be taken proportional to the absolute pressure; hence the increase in density due to a pressure increase of P_{sB} will be $d_{sB} = K_B P_{sB}$. Thus, the increase in weight and the rate of increase in weight of entrapped steam in the exhaust system will be, respectively

$$W_{sB} = V_B K_B P_{sB}$$

and

$$\frac{dW_{sB}}{dt} = V_B K_B \frac{dP_{sB}}{dt}$$

Since the system frequency is assumed constant we can assume that the low-pressure turbines taking steam from the exhaust system of the superposed turbine will have constant governor positions, and therefore constant valve openings. The variation in flow through the low-pressure turbines can then be taken proportional to the variation in pressure of the steam supplied to them, and hence to the variation in pressure in the superposed-turbine exhaust. Consequently

$$z\varphi_v = V_B K_B \frac{dP_{sB}}{dt} + \varphi_{tB}P_{sB} \quad \text{[5b]}$$

Equation [5b] neglects the slight effect of back-pressure changes on the flow through the governing valves of the high-pressure turbine.

Solving the simultaneous differential Equations [1b] to [5b], inclusive, we obtain the following single differential equation with P_{sB} as the variable, representing the transient disturbance

$$\frac{d^3P_{sB}}{dt^3} + \left(\frac{\varphi_{tB}}{V_BK_B} + \frac{a_{x2B}\varphi_{xB}K_y}{K_{xB}a_{y1}^2}\right)\frac{d^2P_{sB}}{dt^2} + \left(\frac{a_{x2B}\varphi_{xB}K_y\varphi_{tB}}{K_{xB}a_{y1}^2V_BK_B} + \frac{a_{x2B}\varphi_{xB}K_y}{K_{xB}a_{y1}^2}\cdot\frac{r_yK_{y1}\varphi_y}{K_ya_{y2}}\right)\frac{dP_{sB}}{dt} + \left(\frac{r_yK_{y1}\varphi_y}{K_ya_{y2}}\cdot\frac{a_{x2B}\varphi_{xB}K_y}{K_{xB}a_{y1}^2}\cdot\frac{\varphi_{tB}}{V_BK_B} + \frac{a_{x2B}\varphi_{xB}K_y}{K_{xB}a_{y1}^2}\cdot\frac{a_{x1B}a_{y1}\varphi_v}{a_{x2B}r_yK_{y1}K_BV_B}\cdot\frac{r_yK_{y1}\varphi_y}{K_ya_{y2}}\right)P_{sB} = 0 \quad \text{[6b]}$$

The following terms can be substituted in this equation:

$$T_{xB} = \frac{K_{xB}a_{y1}^2}{a_{x2B}\varphi_{xB}K_y}$$

is the time constant of the back-pressure regulator

$$T_y = \frac{K_ya_{y2}}{r_yK_{y1}\varphi_y}$$

is the time constant of the governor servomotor

$$T_{IB} = \frac{V_BK_B}{\varphi_{tB}}$$

is the time constant of the exhaust system of the superposed turbine, and

$$T_{MB} = \frac{a_{x1B}a_{y1}\varphi_v}{a_{x2B}r_yK_{y1}K_BV_B}$$

is the response constant of back-pressure governing system. The time constants are a measure of the fastness of action of the different units; the smaller the time constant, the faster the action. The response constant T_{MB} can be explained thus:

Suppose the exhaust pressure is decreased 1 lb per sq in. causing the flow through the superposed turbine to increase by an amount F. If the steam flow F all went into the exhaust system without being able to get out, then the exhaust pressure would rise at a rate (F/V_BK_B) which is equal to the constant T_{MB}. Making the substitutions previously mentioned we obtain

$$\frac{d^3P_{sB}}{dt^3} + \left(\frac{1}{T_{xB}} + \frac{1}{T_{IB}}\right)\frac{d^2P_{sB}}{dt^2} + \left(\frac{1}{T_{xB}}\frac{1}{T_{IB}} + \frac{1}{T_{xB}}\frac{1}{T_y}\right)\frac{dP_{sB}}{dt} + \left(\frac{1}{T_{xB}}\frac{1}{T_y}\frac{1}{T_{IB}} + \frac{T_{MB}}{T_{xB}T_y}\right)P_{sB} = 0 \quad \text{[7b]}$$

The solution of Equation [7b] can be written as $P_{sB} = A_0e^{\alpha_0 t} + A_1e^{\alpha_1 t}\cos(q_1t + \epsilon_1)$. If α_0 and α_1 are each negative, the system is theoretically stable. The constants A_0 and A_1 are determined from initial conditions.

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Supervising Instruments for the 165,000-Kw Turbine at the Richmond Station

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This paper describes the supervisory-control instruments installed on the 165,000-kw turbine at the Richmond Station of the Philadelphia Electric Company. The installation consists of electrical instruments to measure such turbine characteristics as vibration, spindle eccentricity, cylinder expansion, and noise.

The equipment has been in operation for one year, and during this time, the vibrometers and expansion meters have proved to be useful during the starting as well as normal operating periods. The eccentricity meter, although of no great value on a machine equipped with a spindle-turning device, has revealed interesting facts about the movement of the journals in the bearings. The noise-meter transmits the noise-characteristics as heard with a listening rod to a central location on the gage-board. The instrument has not been found very satisfactory because of the great number of noises heard in a machine of this size, making it very difficult to detect a rub whether a listening rod or other means are used for the purpose.

SOME YEARS ago, outdoor turbines were actively discussed as a means of reducing the cost of generating stations, and the development of a group of instruments to indicate and record turbine characteristics was started, in order that the outdoor stations could be operated with minimum attendance.

This development was continued along a somewhat different channel when the Philadelphia Electric Company requested special instruments to safeguard their 165,000-kw Richmond Station turbine, shown in Fig. 1.

The object of the development was to obtain instruments that would quantitatively measure the various factors which determine the safety of starting and operating a turbine.

To any one familiar with power-plant operation, it is a familiar condition that a turbine spindle will bend when the machine has been shut down, because the lower half of the turbine cools quicker than the top half. When the machine is started up again, the spindle is kept rolling for some time at slow speed by passing a small amount of steam through the unit. This small steam flow insures uniform and gradual heating of the spindle and cylinder, and the slow rolling and heating will straighten the rotor shaft if it is bent. If the rotor is brought up to operating speed before it has straightened out, excessive vibration will result and serious rubs may develop.

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Also, if the heating progresses too rapidly, the cylinder will heat faster than the spindle, and the difference in expansion between the two elements may cause mechanical contact in the axial seals in the blade and gland elements.

In recent years, it has been common practice to equip turbines with turning devices that keep the rotor turning at slow speed when the machine is shut down. These devices have materially reduced the heating and rolling period because the rotor shaft is straight when steam is admitted.

From these considerations, it is clear that the characteristics we want to measure during the heating and rolling period are: The eccentricity of the rotor shaft, the expansion of the cylinder, and the noise created by possible rubs between the stationary and rotating elements. When the machine is brought up to speed and during normal operation, we want to measure the vibration.

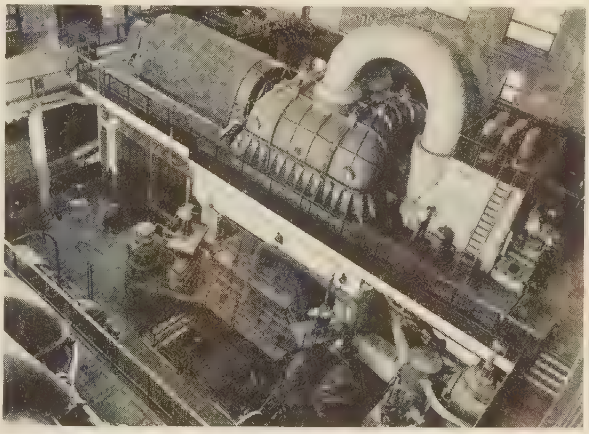

Fig. 1 The 165,000-Kw Turbine Generator at the Richmond Station of the Philadelphia Electric Company

The equipment built for the Richmond turbine consists of the following groups:

1 Eccentricity meters to indicate and record the shaft eccentricity of the high-pressure and low-pressure spindles between the bearing pedestals and the gland cases.

2 Vibrometers to indicate and record the magnitude of vibration in the vertical and horizontal directions at one turbine and one generator bearing.

3 Expansion meter to indicate and record the axial expansion of the turbine cylinder.

4 Noise meter to indicate, through an amplifier system, the noise in the blading and the glands, the indicating meter being supplemented with a pair of headphones to provide an audible signal.

The locations of these instruments are indicated on the turbine cross section shown in Fig. 2. It will be noticed that the vibrometers are located on top of the bearing caps and the eccentricity detectors on the horizontal center line of the shaft. In addition to those shown in Fig. 2, there is a vibrometer on the outboard-generator bearing cap. The noise pickups are located near the glands.

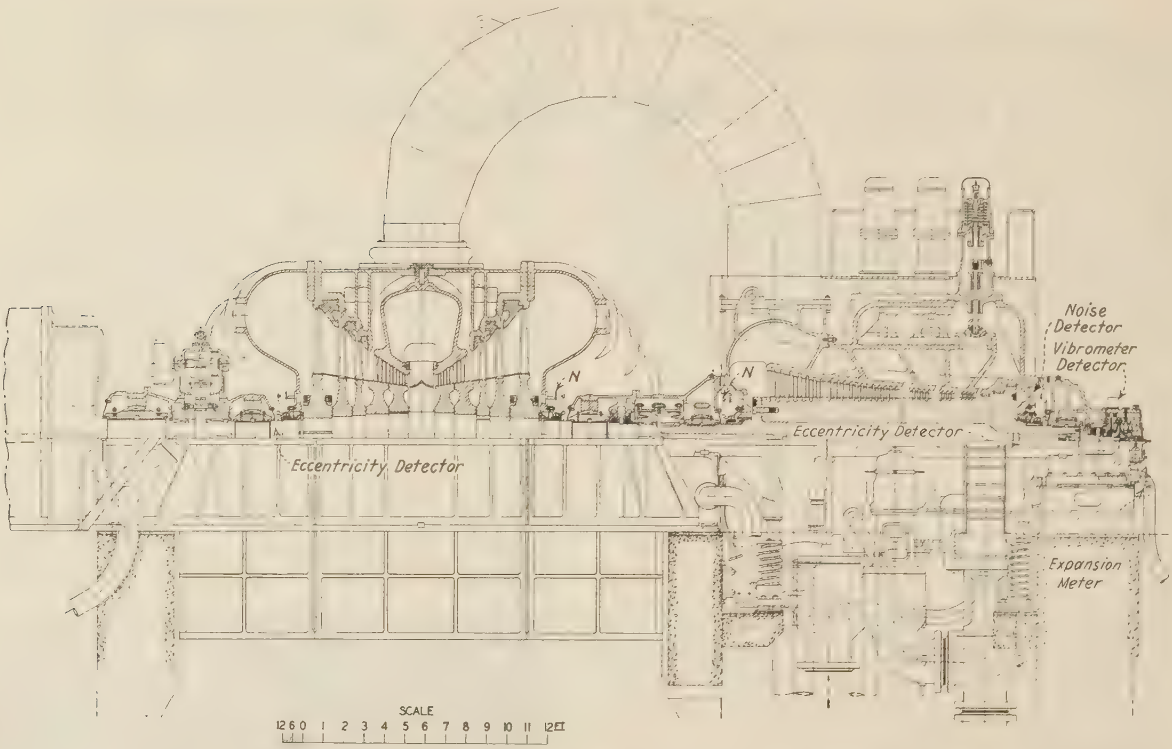

Fig. 2 Cross Section of the Turbine With Location of Instruments Indicated

The expansion meter is located on the front of the thrust pedestal, and the unit is anchored longitudinally at the center of the low-pressure cylinder. Accordingly, the expansion meter measures the longitudinal expansion of the high-pressure cylinder and the expansion of one half of the low-pressure cylinder. The maximum expansion measured is $1^1/_8$ in., at full load, while bleeding for heating feedwater.

The locations of the indicating and recording instruments on the gage board are shown in Fig. 3 as viewed from the throttle valve. These are standard electrical instruments, altered in detail to make them more suitable for the application, and as such require no further comments. The detectors on the turbine, on the other hand, represent some interesting new designs.

Circuits

In the vibrometer and eccentricity meters, a measuring principle used for some time by the Westinghouse Company for measuring small distances has been developed to suit the present application. The same principle is applied to the expansion meter in a slightly modified form to take in the greater displacements. The principle, which has been discussed by C. R. Soderberg,[2] consists of introducing the distance to be measured as a variation in the length of an air gap, controlling the reactance of an inductive circuit. The inductive circuit is excited from an alternating-current source of suitable frequency, and by special coupling of the circuit, its electrical characteristic is a linear function of the length of the air gap. It is an outstanding feature of the equipment designed on this principle that it avoids mechanical contact between moving parts.

In Fig. 4, the eccentricity meter is shown diagrammatically. The detector on the turbine is located between the bearing pedestal and the gland case, and consists of two iron-core transformers T_1 and T_2 mounted as indicated on opposite sides of the shaft. The transformers are identical, and the magnetic circuit for each is made up of the laminated iron core, the air gap between the transformer pole face and the shaft, and a portion of the shaft. The primary windings are connected in series and excited from a constant-voltage a-c source of suitable frequency, while the secondary windings are connected so that the voltages induced will oppose each other and the voltage at the terminals *E-E* will be the secondary differential voltage. With this arrangement, there will be equal voltage drops over each of the primary windings, equal voltages induced in each of the secondary windings, and zero voltage at *E-E* when the air gaps are equal. If the air gaps are not equal, the secondary differential voltage will be a linear function of the difference between the air gaps.

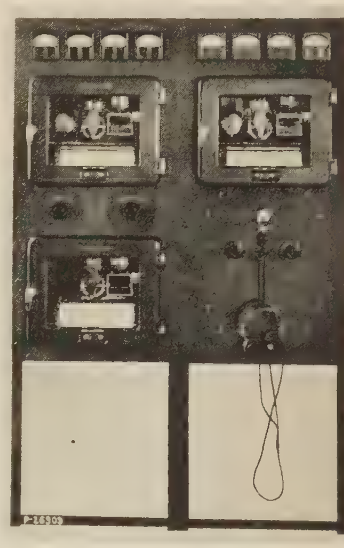

Fig. 3 Location of Indication and Recording Instruments on the Gage Board

When the shaft is bent, it will run eccentric between the two transformers, and each air gap will go through a complete cycle of

[2] "The Vibration Problem in Engineering," by C. R. Soderberg, *Electrical World*, vol. 23, February, 1926, p. 71.

change with a maximum and a minimum value in each revolution. The a-c voltage at the terminals E-E will follow the changes in the air gap and by means of a copper-oxide rectifier, this a-c wave is changed to a unidirectional wave used to actuate the indicating meter and the recorder.

The moving element in the indicating meter, and the galvanometer in the recorder, are of special construction to give a steady reading on the voltage wave for speeds as low as 170 rpm, which is well below the normal turbine heating speed of 200 rpm. When the machine is turning slower than this, the pointer will partly follow the wave, and the eccentricity reading should be taken at the midpoint of the swing of the pointer. When the machine is on the turning gear, the pointer will follow the wave closely and swing from zero to a maximum twice for each revolution. The recorder used is the Leeds & Northrup Micromax recorder of the multipoint type, changing automatically from one detector to another.

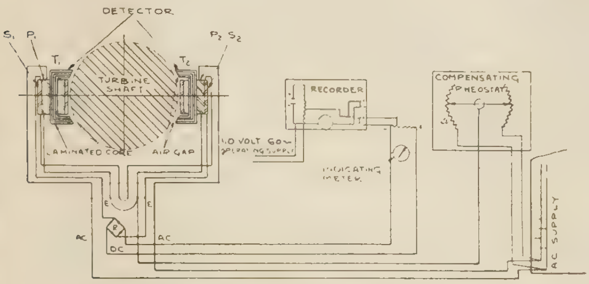

FIG. 4 THE ELECTRIC ECCENTRICITY METER

In the eccentricity meter, this circuit has been supplemented with a compensating rheostat. It is the purpose of this compensating circuit to eliminate the effect of the wedge-shaped oil film between the bearing and the journal that presses the shaft sidewise in the bearings. This sidewise shift results in a difference between the air gaps equal to twice the sidewise movement of the shaft, and even though the shaft may be true, the eccentricity meter will give a reading because the air gaps are not equal. The voltage that produces this reading is a constant one and manual adjustment of this rheostat located on the panel will compensate for the difference in air gaps and give a zero reading if the shaft is true. If the shaft is not true, the rheostat is adjusted to give a minimum reading on the meter. This minimum reading corresponds to the wave portion of the voltage only, as caused by the changes in the air gaps. And thus, in spite of the movement in the bearings, a true reading of the eccentricity may be obtained.

The vibrometer is based on the same measuring principle as the eccentricity meter, and in each vibrometer there is one measuring circuit for horizontal vibration and one for vertical vibration.

Fig. 5 shows two vibration detectors mounted for testing. One unit is shown complete with cover, while the other is shown with the cover removed. The overall dimensions are approximately 7 × 7 × 7 in., and the instrument is conveniently located on the bearing cover. The mechanical design is similar to an amplitude meter built on the seismic principle. The weight W, which includes the four transformers T, is supported flexibly by a cantilever beam, having equal flexibility in the vertical and horizontal directions. The air gaps are formed between the transformer cores and the laminated-pole shoes A, which are rigidly attached to the frame. When vibrations are set up in the bearing pedestal supporting the frame, the pole shoes will follow the vibration, while the flexibly supported transformers will remain fixed, due to the inertia of the weight. Thus, there will be a relative motion between the transformers and the pole shoes, varying the length of the air gaps by an amount equal to the range of the vibration. The two vertical and the two horizontal transformers are connected so that each pair forms an independent measuring circuit, and the instrument will give simultaneous readings of horizontal and vertical vibration.

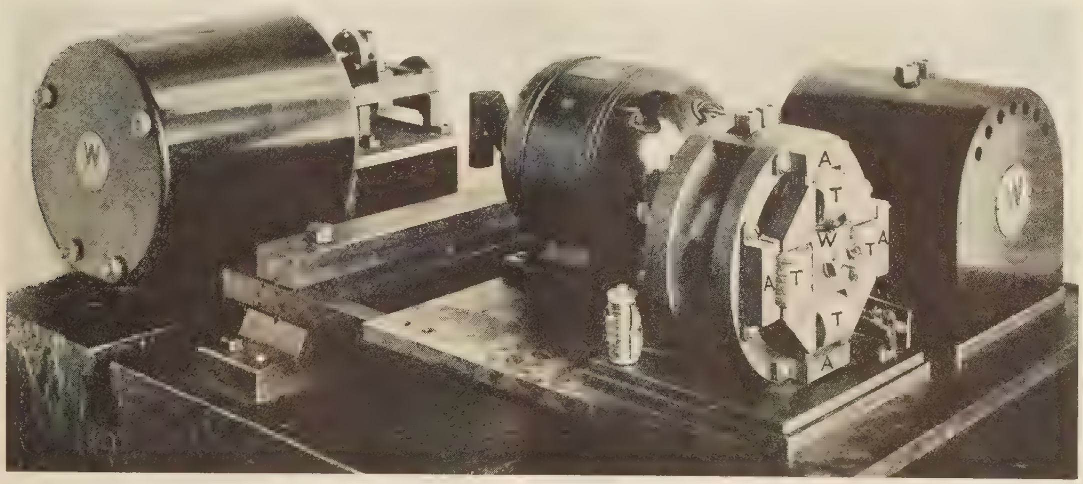

FIG. 5 TWO VIBRATION DETECTORS MOUNTED FOR TESTING

A sectional view of the expansion detector is shown in Fig. 6. The primary coil is attached to the pedestal and rides on the stationary laminated armature C. The section B of this armature serves as core for the secondary windings and, as the sliding pedestal moves the primary from left to right, the voltage induced in the secondary will increase. With this arrangement, the voltage increase in the secondary is a linear function of the movement of the primary. The alternating voltage induced in the secondary is rectified, as in the case of the instruments previously discussed, and the rectified current actuates the indicating and recording instruments.

The noise meter consists of detectors mounted on the turbine, and connected to a suitable amplifier with a calibrated output meter and headphones, located on the gage board. By means of a selector switch, any one of the pickups as desired may be connected to the amplifier and the noise at that location checked.

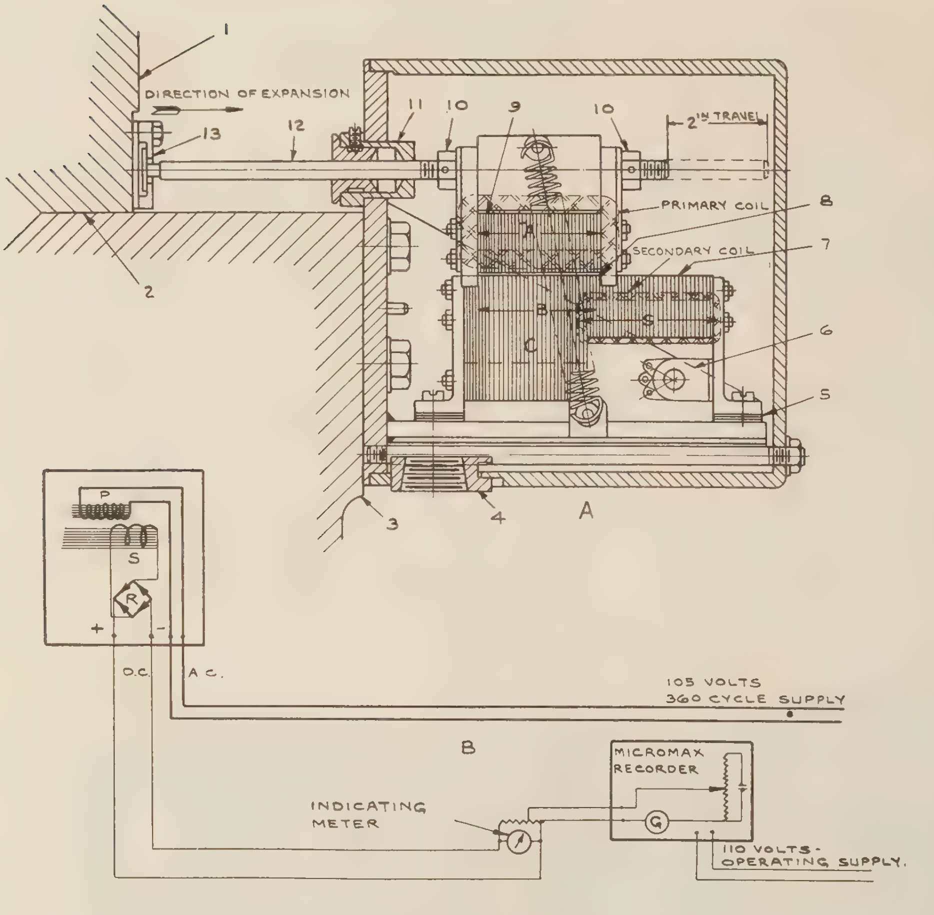

Fig. 6 The Expansion Detector

The amplifier is also equipped with an attenuator so that the volume can be adjusted to suit the individual observer.

The detector is a high-frequency magnetic-vibration pickup, the diagram of which is shown in Fig. 7, and consists of a permanent magnet *M* flexibly supported, and a coil *C* mounted between the magnet-pole faces and rigidly attached to the frame. When the case vibrates, the magnet will remain stationary due to its inertia, while the coil will vibrate with the frame and, due to the relative motion between the magnet and the coil, a voltage will be induced in the coil.

The sound waves are transmitted through the cylinder as high-frequency waves and are picked up by the detector in the same manner as a listening rod picks it up, and results in relative motion between the magnet and the coil. The magnitude and wave form of the voltage is directly related to the vibration, and the noises heard in the headphones have the same characteristics as those heard with a listening rod.

Operation

When the turbine was put in service, it was found that with a turning device on the machine, there was no great need for an eccentricity meter. The spindle was generally straight enough so as not to require any further attention.

The instrument itself brought out some very interesting points. The spindle moved around in the bearings considerably more than anticipated. As the speed was increased from slow rolling, there was a uniform movement of the rotor shaft toward the left-hand side of the machine. The compensating rheostat had to be used continually, and the record was of little value except at the points where it had been compensated.

When the oil temperature came up, there was a tendency for the spindle to move back toward the center, and as load was applied, the movement toward the center continued. However, the journal never returned to its standstill position or turning gear position in the bearings.

It was also found that the readings were relative only, because of the nonuniform magnetic characteristics of the surface of the forged shaft. Because of this nonuniform characteristic, it was not possible to balance the circuit. There is also a certain amount of residual magnetism in the shaft that gives trouble, and on the high-pressure end, the continual high temperature has resulted in insulation trouble.

The first cure considered for these troubles was to install a laminated ring on the shaft, move the detector to the other side of the bearing, and measure what might be termed reverse eccentricity. However, investigations on other machines made in connection with general balancing problems, and investigations of shaft vibrations by the use of a stroboscope show that the eccentricity measured on each side of a bearing is not the same, except at very low speeds. The fulcrum does not stay constant at the center of the bearing, but moves out with increasing amplitude and speed. Thus, the reverse eccentricity will decrease as the actual bow in the shaft increases, pass through zero and then

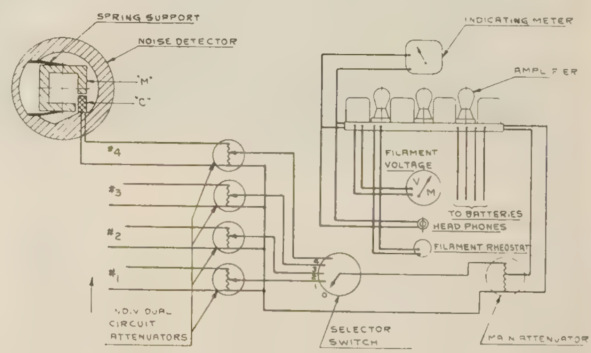

Fig. 7 Diagram of High-Frequency Magnetic-Vibration Pickup

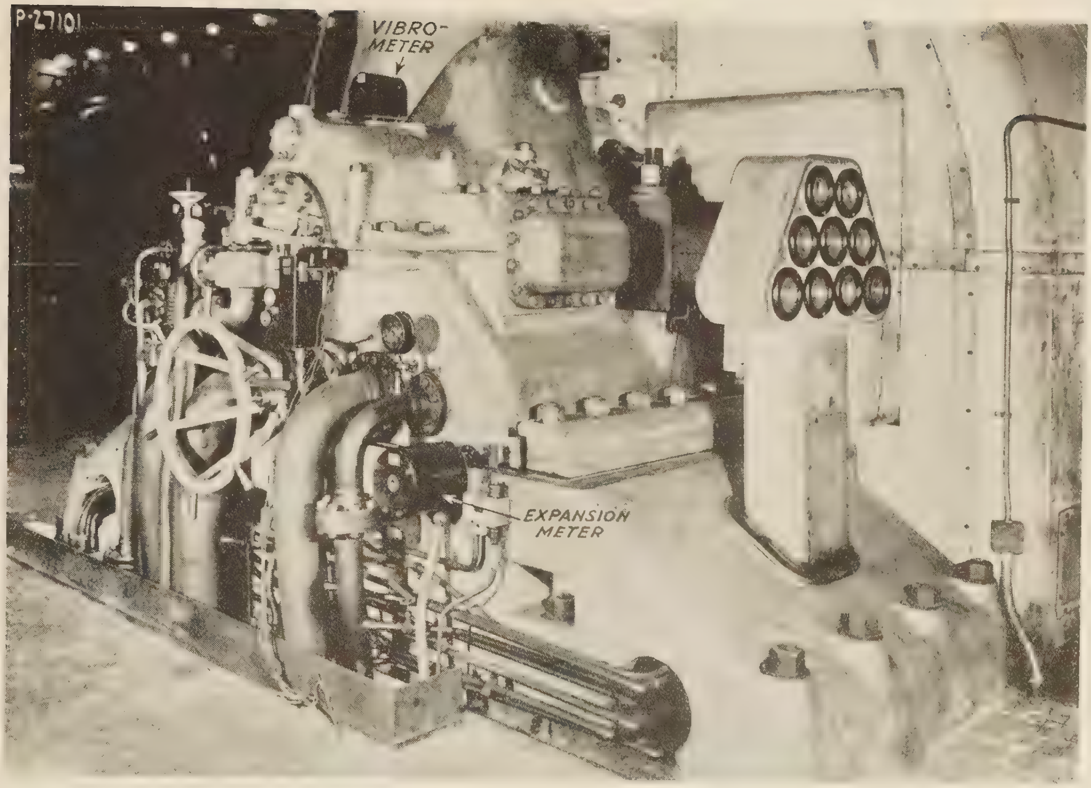

Fig. 8 The Thrust Pedestal With Vibrometer and Expansion-Meter Locations Indicated

.gain increase. On three bearing units with solid coupling, these onditions have been found to be quite complicated. The am- ılitudes and phase relations on each side of the middle bearing lepend upon the critical deflection curve the rotor may happen o be in, and the location of the nodes.

On the basis of these findings, the use of reverse eccentricity vas abandoned, with the belief that, in view of the slight need or an eccentricity meter on a machine equipped with a turning ear, the most satisfactory solution from an economic, as well as ervice standpoint, will be to use a carbon-brush rider on the shaft nd lift the rider off the shaft when not in use. Such a rider can e built to follow the shaft at all speeds up to running speed. 'he instrument then would be indicating only and switched on nd off as desired. At the present time, the eccentricity re- order is not in use, but the indicating meters are used as a rela- ve indication of eccentricity.

The vibrometer has been found to be of real value and very ttle trouble has been experienced with this unit. When the nit was put in service, the heat from the bearing cap caused some xpansion trouble, but this has been overcome by separating the ase from the bearing cap with an air space. The arrangement for one four-point recorder, of the print-wheel type to record ne vibrations vertical and horizontal at each of two bearings; nd four indicating meters, one for each of the vibrations. Fig. 8 nows the thrust pedestal, with the locations of the vibrometer nd expansion meter indicated.

Up to this time, the vibrometer has been most useful when inging the machine up to speed. A turbine-generator unit of is size has several critical speeds below the running speed, some which originate in the foundations. With the indicating eters located as they are on the gage board, the operator at the rottle is in a position to keep an eye on the meters and bring the achine through the critical speeds quickly and with a minimum of vibration. During normal operation, slight vibration changes caused by load changes are immediately indicated by the meters and, if unusual vibration should develop, the record will show the exact time of its inception and make it possible to tie it in with other operating changes.

Fig. 9 shows a record of the vibrations experienced when putting the unit back in service after an overnight shutdown, during which it was kept on the turning gear.

The actual records are stamped on the chart by the recording instrument and consist of a series of groups of dots registered at approximately 2-min intervals, each dot having beside it an identifying number so that the horizontal and vertical records at the two ends of the shaft can be distinguished.

This record, while clearly legible on the chart itself, is not suitable for reproduction. Therefore, for convenience, the points on Fig. 9 have been transferred to separate arbitrary base lines and connected by light lines. No. 1 and No. 2 base lines show vertical and horizontal vibrations, respectively, at No. 1 bearing, while base lines Nos. 3 and 4 show vertical and horizontal vibrations, respectively, at the outboard-generator bearing. It may be noted base line No. 4 remains on zero throughout. Such vibrations as were noted were of such small magnitude that they cannot be shown on this chart. This record runs from 4:30 a.m. to 8:45 a.m. and the points at which the unit was rolled by steam, put on the line, and brought up to 130,000 kw are marked A, B, and C, respectively. This will permit the vibration record to be compared with the expansion record shown in Fig. 10 and referred to later.

There has been occasion to try the vibrometer on other turbines in the field where there were vibration problems, and it proved itself to be a valuable tool. On one machine where it was used, the machine had developed spells of vibration the period and amplitude of which were accurately recorded on the meters.

On the basis of these field experiences, as well as that at Richmond Station, the author believes that the continuous log of vibration obtained by the recorder is a valuable operating record that justifies its more general use.

The vibrometers also have proved to be of rugged construction and able to withstand moving around from one job to another. When the detector came back from the field job, it was set up in the laboratory to check, and found to be in perfect balance ready to be used again without readjustment.

Fig. 10 shows a record of the expansion while putting the unit in service after an overnight shutdown on the turning gear and corresponds, over the period from 4:30 a.m. to 8:45 a.m. to the vibration record shown in Fig. 9.

At 4:50 a.m. steam was admitted to roll the unit, thus taking it off the turning gear. At 5:15 a.m. acceleration was begun and the vacuum was gradually increased to its normal value. At 6 a.m., point B, the machine was up to speed and was put on the line, and at 8:00 a.m. the unit was carrying 130,000 kw. This record shows the temporary cooling effect of rolling the unit under steam with increasing vacuum, after it has been rolled for several hours on the turning gear with low vacuum and condensate recirculation from the air ejectors.

It also shows the time lag in the expansion since, though the unit was brought up to 130,000 kw at 8:00 a.m. and kept there, the unit was still expanding at 8:45.

The small vertical marks noticeable at 4:58, 5:40, 6:22, 7:05, and 7:47 a.m., represent the automatic checking of the instrument against a standard cell.

The operating experience with the expansion meter has been entirely satisfactory. The instrument draws a line record of the thermal expansion and contraction of the cylinder as it heats and cools, due to starting and stopping. It also shows the effect of load changes on the machine.

The information given by the expansion meter has proved to be of great assistance during the starting period, in that it shows accurately whether or not the cylinder parts are expanding normally and gives warning if the sliding joint is not functioning properly. The instrument readily shows the difference between greasing the pedestal and letting it run dry. The dry pedestal shows up as a jagged line on the chart. The general effect of such restricted expansion is to twist the cylinder and since the cylinder is keyed to the bearing pedestal, it will throw the bearings out of line and vibration will set in. It is possible that such twisting of the cylinder may be sufficient to cause a rub in the machine.

In addition to the vibrometer, the expansion meter has been used on several other turbines in the field. On one of these, the instrument proved itself extremely useful, by calling attention to an abnormal contraction of the cylinder at a certain load. This contraction was found to be caused by local cooling of a section of the cylinder by gland sealing water, and proper steps to correct the trouble were taken.

The generally accepted method of detecting rubs in a turbine, caused by mechanical contact between the stationary and moving parts of the sealing details in the glands, or blading, is to use a listening rod. The noise meter gives results similar to the listening rod and has the advantage that it brings the noise to one central location at the gage board. As is the case with the listening rod, the operator has to get accustomed to the various noises in the machine before they are able to detect a rub.

On a large machine, it is extremely difficult to separate the various noises heard in the machine, and the steam noise tends to drown out any rub that might take place. On large machines, the operators seldom attempt to use a listening rod after there is any appreciable steam flow through the unit.

On smaller units, say 10,000 to 15,000-kw, the various noises are more readily distinguished, and the noise meter has been used on some of these smaller units with a considerable degree of success.

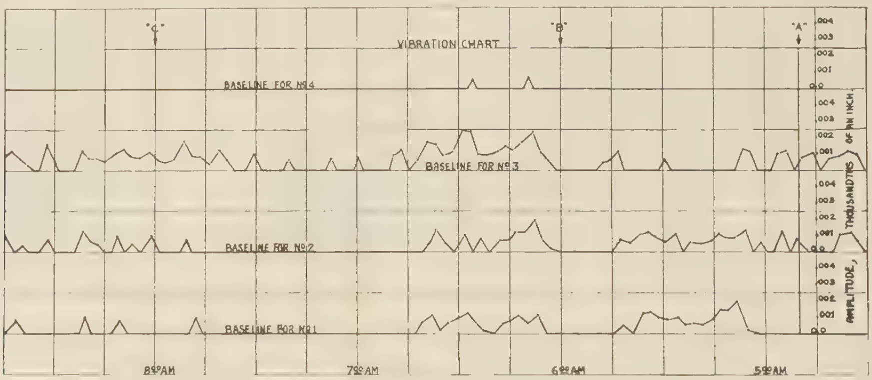

FIG. 9 VIBRATION CHARACTERISTICS WHEN PUTTING THE UNIT BACK IN SERVICE AFTER AN OVERNIGHT SHUTDOWN

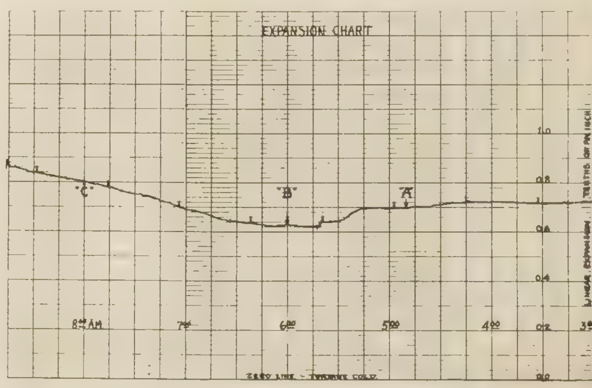

FIG. 10 EXPANSION RECORD CORRESPONDING TO VIBRATION RECORD SHOWN IN FIG. 9

Consideration has been given to supplementing the amplifier with a filter that would eliminate the steam noise. The frequency of the steam noise, however, varies with the valve lift and the filter could not eliminate the high-pitched steam noises sometimes encountered without also filtering out a rub. The filter, however, would have the advantage in that all low-frequency noise would be eliminated, such as bearing and pump noise, and the noise from the circulating pumps transmitted through the condenser to the cylinder is of considerable volume. Therefore, the elimination of these deeper frequencies of the noise might be a great improvement.

Tests of a 50,000-Sq Ft Surface Condenser at Widely Varying Temperatures, Velocities of Inlet Water, and Loads

By G. H. Van Hengel,[1] Detroit, Mich.

The purpose of this paper is to enhance the understanding of the undercooling of condensate and of the heat-transfer coefficient. The author replaces the heat-transfer coefficient with a simplified form of the reciprocal of the heat-transfer coefficient; viz., the local resistivity taken at the cooling-water-inlet end of the condenser. Exhaustive tests on a modern condenser show first, that superheating of the condensate, as generally defined, is possible if provision is made for the recovery of the velocity energy of the steam flowing into the condenser; and second, that a practicable correlation exists between the resistivity at the cooling-water-entrance end of the condenser and the Reynolds number of the cooling water at the tube entrance.

Surface condensers have been in use for many years, but little accurate information is available on their actual performance under different operating conditions. For the modern power station, with large steam turbines, the condenser is a major economic problem. More knowledge as to condenser performance is needed in order to strike the best balance between the large space the condenser occupies, the great quantities of circulating water it requires, the power it consumes for auxiliary units, and the low back pressures demanded to reduce the steam rate of the turbine. It is for this reason that the author, assisted by an improved testing technique and a more detailed analysis, has prepared this paper.

For a thorough understanding of condenser performance, it is necessary to consider two different circuits which involve three different fluids, namely, steam, air, and water. These two circuits are:

(*A*1) The steam-air circuit, which changes principally with different loads on the turbine or with different amounts of air in-leakage to the condenser.

(*A*2) The hydraulic circuit, which varies principally with different speeds of the centrifugal pumps, which are called circulators in condenser practice.

Furthermore, from an economic point of view two resistances are to be included. These are:

(*B*1) The resistance to heat flow, which can be split into (*a*) the resistance at the inside of the tube in the water space, (*b*) the resistance through the wall of the tube, and (*c*) the resistance outside the tubes in the steam space.

(*B*2) The resistance to the fluid flow in the circuits mentioned under (*A*1) and (*A*2). This can be divided into (*a*) the resistance to the steam flow outside the tube banks, through the tube bank, and through the air cooler; and (*b*) the resistance of the cooling-water flow through tubes, the water boxes, and piping.

This paper deals with the parts of the steam-air circuit outside the tube banks. It also deals with the resistance to heat flow, in which a theoretical analysis of the three component resistances is introduced in order that only the total resistance need be considered in the practical application. The hydraulic circuit is not treated in this paper.

Two methods can be followed in analyzing a condenser. One method is to study the performance of the condenser as a whole, and the other is to study each tube individually. The author has preferred to take the point of view of analyzing first the condenser as a whole or as an assembly of parts so that, later on, a study of the tubes themselves can be made to improve the performance.

To furnish the material for the analysis of a condenser, the author ran exhaustive tests for nearly one year on one of the newest condensers installed by The Detroit Edison Company. The test conditions were varied over five circulating-water inlet temperatures ranging from 32 to 75 F, over four generator loads ranging from 20,000 to 50,000 kw, and over nearly a full range of circulator speeds for one and two pumps, corresponding to a range of water velocities from 3 to 7 fps through the condenser tubes.

Furthermore, it is believed that the accuracy of the data obtained is noteworthy. In testing, an attempt has been made to keep the accuracy of the measurement of the steam and water temperatures within 0.1 F, and the absolute pressures within 0.003 in. hg. In a modern, single-pass condenser, where the pressure drop through the whole condenser is as low as 0.015 in. hg, and the temperature rise of the circulating water is as low as 4 F, high precision of the absolute values of pressure and temperature is a necessity.

In order to give a better understanding of the paper, a description of the unit is given first with a tabulated summary of data and results for some representative tests.

The main subjects of this paper are discussed in the analysis and are limited to (*a*) the undercooling of the condensate, and (*b*) a method of representation of the all-year-around performance of a condenser by a single curve. Only the results of the tests on the condenser referred to previously have been taken to discuss these subjects, although many additional proofs from tests on other condensers could have been added. The method of analysis used for this condenser is applicable to many others which show

[1] Consultant, Detroit Edison Company. Mem. A.S.M.E. Mr. Van Hengel was graduated from the Technical University at Delft, The Netherlands, in 1918, with an M.E. degree. After graduation he spent three years in Holland on design and construction of industrial plants. In 1921 he came to the United States for a study trip and for half a year visited all kinds of factories. He spent six years with the Worthington Pump and Machinery Corporation, Cincinnati, Ohio, and Harrison, N. J., on design, testing, and research of centrifugal pumps, condensers, dry-vacuum pumps, and steam air ejectors. For two and one-half years he was chief engineer of the Issoudun Aircraft Manufacturing Company, Northville, Mich., makers of all-metal amphibians. Since 1930 he has been a consultant for the Detroit Edison Company on steam condensers and auxiliaries.

Contributed by the Power Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

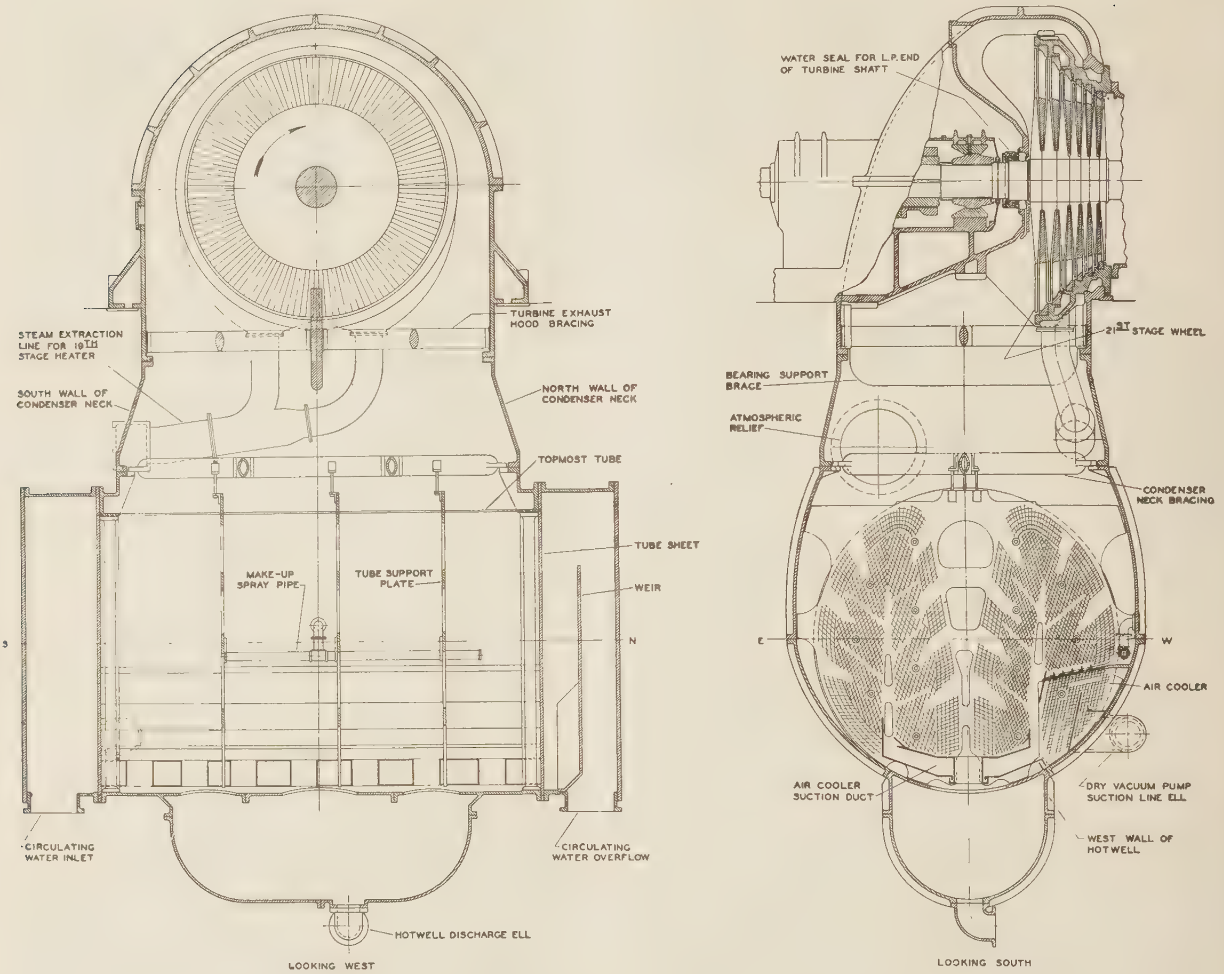

Fig. 1 Sectional Elevations of the Condenser and Turbine Exhaust

the same characteristics, although in varying degrees according to their features of design and range of operating conditions.

An appendix concludes the paper by giving some examples of the accuracy of the test instruments and the data obtained.

DESCRIPTION OF THE UNIT

The unit consists of the turbogenerator, condenser with internal air cooler and reciprocating dry-vacuum pump, two circulating pumps, generator air cooler, and four extraction feedwater heaters. The turbine is at the west end of the turbogenerator east-west axis, and the condenser axis is transverse to this axis, the circulating pumps being south of the condenser. The unit, which is the No. 13 main unit in Delray power house No. 3, was placed in service late in 1933.

The Condenser

This is a single-pass, Worthington condenser, with an external cooling surface of 50,630 sq ft consisting of 8143 brass tubes including 698 tubes segregated in an internal air-cooler located on the lower west side of the condenser as shown in Fig. 1. The tubes are 1 in. outside diameter, have a wall thickness of No. 18 Bwg or 0.049 in., and a length of 23 ft 9 in. between the tube sheets.

Steam enters the top of the condenser through a 14 × 21-ft neck, the net flow area at the braces being approximately 250 sq ft. This steam has direct access to the hotwell by two paths, one of which is through a lane between the two folded-layer tube banks and holes through the air-cooler suction duct below. The other path is between the east tube bank and the east wall of the condenser shell.

The air-cooler suction duct enables air to be drawn from the center of the east tube bank as well as from the center of the west tube bank. A vertical condensate baffle in the air cooler creates a two-pass effect. The air is taken off at the inlet-water end of the condenser shell at the bottom of the air box through a 16-in. pipe which leads to the suction of the two-stage feather-valve 35 × 18-in. reciprocating dry-vacuum pump.

The Turbogenerator

This unit is capable of an output of 50,000 kw at back pressures under 2 in. hg abs, with initial steam conditions of 375 lb per sq in. gage and 700 F, at a speed of 1200 rpm. The 21-stage Curtis turbine has a by-pass admission valve to the fifth wheel which is opened only after the governor control valve on the main line to the bowl is wide open, which occurs at a load of approximately 43,000 kw. Steam is extracted for feedwater heating at the 9th, 13th, 16th, and 19th wheels. The 21st wheel has a diameter of 13 ft 11 in. across the blade tips; it has 372 blades 30.25 in. high across the free-flow area with an exit angle of 45 deg.

Steam escaping through the high-pressure labyrinth shaft packing is condensed in the 16th-stage feedwater heater. The heater drains cascade. The 19th-stage heater drains, which therefore consist of all bled steam and the high-pressure shaft packing steam leakage, are pumped into the feedwater line between the 16th- and 19th-stage heaters, and this total flow of feedwater is raised to a pressure of about 500 lb per sq in. gage by the boiler feed pump between the 9th- and 13th-stage heaters.

TESTING PROCEDURE AND TECHNIQUE

Operating Conditions for Test

A total of 87 runs was made with bleeding operation of the turbine and one with nonbleeding operation. At each different water-inlet temperature, combinations were made of four generator-load conditions from 20,000 to 50,000 kw and of four circulator speeds from the lowest of 180 rpm to the highest of 240 rpm. The two circulators were always operated at equal speeds. To obtain low velocities of the cooling water, 27 runs were made with only one circulator operating. The dry-vacuum pump speed was always set at the usual operating speed of 100 rpm.

Generally four, 1.5-hr runs were made in one day with intervals of 1 hr, readings being taken at 15-min intervals. Preliminary readings were taken during the half hour preceding a run. During any one day, the turbine load was not changed with the exception that runs at loads of 30,000 and 40,000 kw were made on the same day.

In order to have constant steam pressures in the condenser and constant outlet temperatures of the circulating water, the constancy of the steam flow to the condenser was given special attention. This flow is the difference between the total steam flow to the turbine at its throttle and the extraction steam flow to the feedwater heaters. The total steam flow to the turbine was kept constant by locking its governor and by maintaining the station steam pressure exceptionally steady. The extraction steam flow was held constant by maintaining a steady feedwater flow through the heaters of this unit. This was done by close regulation of the several feedwater pumps of the station, which operate in parallel.

As the station was under base load during test periods, good results were obtained by this procedure. For 57 runs the fluctuations of the back pressure during the run were within 0.010 in. Hg, as measured at the south wall of the condenser neck; and for 31 of these 57 runs, these fluctuations were within 0.005 in. Hg. The air leakage was very small, varying between 0.7 and 2.0 cfm during all tests with the exception of the one nonbleeding run No. 36 when air was admitted to the dry-vacuum pump suction to obtain 1 in. Hg back pressure.

General Testing Technique

Many more observations were made than the minimum required for the calculation of the results. In this way the data could be strengthened greatly by cross checks; that is, absolute checks by measurements of the same quantity by separate observations using instruments operating on different principles, and relative checks by using known relationships between different measured quantities.

For each of these condenser tests, about 170 different items were measured. As the moisture content and flow of the wet steam at the condenser entrance cannot be measured directly with satisfactory accuracy, about 60 of the items were measured on the turbogenerator and the feedwater heaters in order to evaluate these wet-steam conditions. For the condenser steam circuit and heat transfer, 40 temperatures and eight steam pressures were measured. In addition, circulating-water pressures were measured at 11 points and water levels at five points in order to calculate hydraulic resistances and circulating-pump performance.

Table 1 gives the principal data and calculated results for 30 representative runs. Fig. 2 shows the locations of many of the temperature and pressure measurements. A complete account of all instruments would be tedious, but some details concerning a few of the outstanding measurements are given in the Appendix.

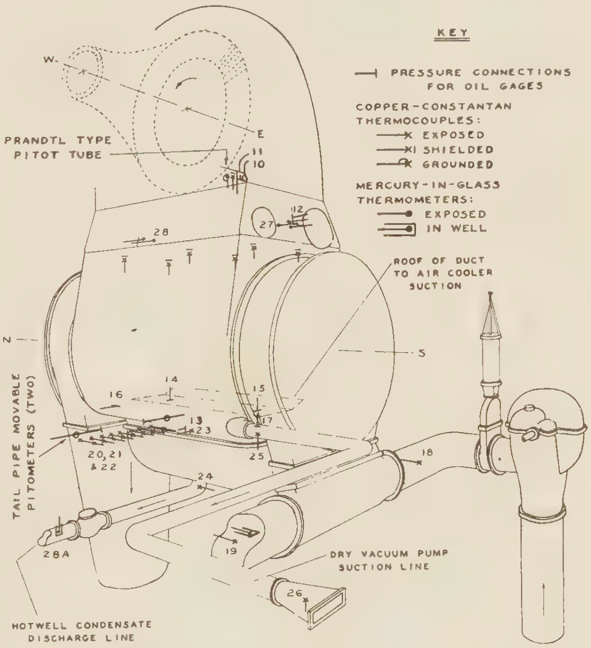

Fig. 2 Location of Principal Condenser Measurements
(The numbers refer to the items listed in Table 1.)

ANALYSIS

(A1) The Steam-Air Circuit

One of the circuits which has been the least mentioned in condenser design is the steam-air circuit. This circuit is the most difficult to analyze. The author has found that two items help in the study of this circuit problem. These are: (1) Measuring the exhaust velocity and static pressure of the steam coming out of the turbine, and (2) measuring the successive steps of pressure drop through the condenser. The first item is in reality a turbine problem,[2] and it will be mentioned only in so far as its results are necessary in considering the condenser problem. The second item is a highly essential one in analyzing a condenser, because the pressure drop through a condenser should be kept as small as possible in order to obtain the best performance.

The steam-air circuit can be divided into three parts: (1) The steam flow outside the tube banks, (2) the steam flow through the tube banks, and (3) the steam flow through the air-cooler. In this paper only the steam flow outside the tubes will be handled, and this subject will deal with the steam entering the condenser and the superheating of the hotwell condensate.

The Steam Flow Entering the Condenser. From an examination of the steam circuit it is evident that the steam leaves the last wheel of the turbine at widely varying velocities, depending upon

[2] "Steam Turbine Testing," by C. H. Berry, *Mechanical Engineering*, vol. 57, 1935, pp. 705–709.

TABLE 1 SUMMARY OF

Item												
1.	Run No.	65^{s}	66^{s}	67^{s}	68	72^{s}	73^{s}	76^{s}	79^{s}	81^{s}	88	2^{s}
2.	Date[a] (duration 1-1/2 hour in most cases)	1/7	1/7	1/7	1/7	1/8	1/9	1/9	1/10	1/14	1/15	3/26
3.	Average inlet temperature of circulating water, F	32.97	33.02	33.21	33.28	32.82	33.17	33.04	33.15	32.75	33.48	42.08
4.	Load at generator terminals by rotating standard, kwhr/hr	49680	49720	49910	49640	50030	40200	40530	29960	20540	20640	50520
5.	Circulators running	2	2	2	2	West	2	West	West	2	West	2
6.	Average speed of circulators, rpm	237.4	221.2	200.7	178.2	178.2	239.7	178.9	240.5	239.5	178.6	220.3
7.	Total d-c power input to circulating-pump motors, kw	617	496	366	259	137	630	139	335	637	137	498
8.	Speed of dry-vacuum pump, rpm	100.0	100.0	99.4	99.0	99.5	99.5	99.8	99.5	104.0	104.0	98.9
9.	Dry air flow at vacuum-pump discharge by gasometer, std cu ft/min[b]	1.210	1.202	1.216	1.250	1.023	0.998	0.875	0.844	0.960	0.913	1.063
	Exhaust-Steam Pressures by Absolute-Pressure Oil-Gages											
10.	Prandtl tube total, uncorrected for attack angle, in. Hg at 32F	0.901	0.907	0.910	0.997	0.915	0.713	0.729	0.543	0.399	0.436	0.857
11.	Prandtl tube static, uncorrected for attack angle, in. Hg at 32F	0.4[illegible]	0.452	0.442	0.475	0.606	0.356	0.485	0.349	0.271	0.334	0.554
12.	South wall of condenser neck at N-S centerline, in. Hg at 32F	0.4[illegible]	0.440	0.463	0.489	0.631	0.371	0.516	0.370	0.285	0.350	0.607
13.	West wall of top of hotwell at E-W centerline, in. Hg at 32F	0.548	0.561	0.579	0.597	-- [c]	0.457	0.573	0.414	0.312	0.371	0.694
14.	Roof of air-cooler suction duct at north end, in. Hg at 32F	0.381	0.391	0.41[illegible]	0.439	0.580	0.333	0.473	0.339	0.267	0.327	0.568
15.	Roof of air-cooler suction duct at south end, in. Hg at 32F	[illegible]	0.364	0.384	0.408	0.539	0.317	0.444	0.324	0.264	0.314	0.534
16.	Air-cooler offtake at north end, in. Hg at 32F	0.282	0.283	0.300	0.301	0.383	0.289	0.335	0.278	0.264	0.279	..
17.	Condenser end of vacuum-pump suction line, in. Hg at 32F	0.273	0.270	0.245	0.287	0.344	0.275	0.305	0.267	0.262	0.263	0.466
	Principal Temperatures by Copper-Constantan Thermocouples											
18.	Inlet circulating water at east pump discharge, F	32.98	33.04	33.23	33.30	..	33.18	..	..	32.76	..	42.08
19.	Inlet circulating water at west pump discharge, F	32.96	32.99	33.18	33.25	32.82	33.15	33.04	33.15	32.73	33.48	42.07
20.	Outlet circ water 8-point average for east half of tail pipe, F	39.71	40.34	41.40	42.64	47.88	38.52	45.08	39.74	35.90	40.30	49.19
21.	Outlet circ water 8-point average for west half of tail pipe, F	38.82	39.36	40.35	41.56	47.24	37.75	44.49	39.23	35.24	39.98	48.60
22.	Outlet circ water 16-point average for tail pipe top, F	39.26	39.85	40.87	42.10	47.56	38.14	44.79	39.48	35.57	40.14	48.90
23.	Steam at west wall of top of hotwell at E-W centerline, F	61.53	61.85	62.81	63.76	68.79	56.29	62.59	53.50	46.03	51.00	67.77
24.	Condensate in discharge line ell at bottom of hotwell, F	60.44	60.98	61.95	62.83	68.13	55.48	62.01	52.88	45.46	50.26	67.30
25.	Air-steam mixture in condenser end of vacuum-pump suction line, F	49.00	48.67	48.87	49.18	53.11	49.41	51.50	49.94	49.51	51.12	61.20
26.	Air-steam mixture in pump end of vacuum-pump suction line, F	75.25	73.55	74.05	75.15	77.00	77.66	78.61	75.80	73.68	80.16	77.50
	Principal Temperatures by Mercury-in-Glass Thermometers											
27.	Steam 9 or 10 inches in from south wall of condenser neck, F	5[illegible].15	[illegible].77	5[illegible].12	56.65	64.38	49.14	58.78	49.52	42.99	48.74	62.76
28.	Steam 9 or 10 inches in from north wall of condenser neck, F	51.[illegible]	5[illegible]	[illegible].81	55.88	63.93	48.25	58.39	49.26	42.53	48.39	62.41
28A.	Well in hotwell-condensate discharge line after check valve, F	60.4[illegible]	61.25	6[illegible].19	63.06	67.73	55.83	61.69	53.13	45.65	50.35	67.44
29.	Room at south wall of condenser neck, F	7[illegible].4	75.7	77.6	78.3	81.9	77.3	78.6	77.9	75.6	82.2	81.6
30.	Room at north wall of condenser neck, F	90.7	91.7	92.6	93.4	96.9	91.9	93.7	92.8	90.7	96.1	93.4
31.	Room at west wall of hotwell, F	67.2	66.8	68.1	68.6	71.9	69.0	69.1	68.7	64.9	71.7	73.5
32.	Room at condenser end of vacuum-pump suction line, F	72.1	73.3	71.0	73.7	75.2	73.6	73.0	74.0	68.9	80.0	80.8
33.	Room at pump end of vacuum-pump suction line, F	77.8	77.5	78.6	79.0	81.5	80.9	80.2	81.8	78.9	86.7	81.9
	Principal Turbine Data and Calculated Results											
34.	Generator power factor, per cent	74.6	74.6	75.6	75.8	75.4	74.8	76.8	78.7	76.5	79.6	75.3
35.	Turbo-generator losses including 400-kw mechanical losses, kw	1790	1790	1790	1780	1790	1680	1680	1570	1510	1510	1800
36.	Turbine internal load at blades, kw	51470	51510	51700	51420	51820	41880	42210	31530	22050	22150	52320
37.	Output of all extracted steam (per heat balance), kw	6180	6120	6060	5870	5860	5640	5500	3860	2490	2430	6010
38.	Output from steam going to condenser, kw	45290	45390	45640	45550	45960	36240	36710	27670	19560	19720	46310
39.	Steam flow at throttle calc from boiler-feed venturi meter, lb/hr	495300	494600	497000	491300	490800	391400	389500	287300	205500	204200	498100
40.	Steam admission through overload by-pass line to 5th wheel, lb/hr	83800	84600	89000	91400	82300	0	0	0	0	0	75700
41.	High-pressure shaft packing steam leakage, lb/hr	4390	4510	4430	4870	4670	4220	4260	3280	3040	2780	3830
42.	Total steam extraction (by heat bal of feedwater heaters), lb/hr	115200	114700	115300	112500	110300	86500	83900	58400	38600	37500	115500
43.	Wet steam flow to condenser from 21st wheel of turbine, lb/hr	375600	375400	377400	373900	375900	300700	301300	225500	163800	163900	378700
44.	Energy utilized by blades from steam going to condenser, Btu/lb	411.4	412.5	412.6	415.6	417.1	411.2	415.7	418.6	407.3	410.5	417.2
45.	Well thermocouple[d] in steam line before throttle, F	719.9	717.7	714.8	722.6	736.4	721.5	739.5	740.9	712.6	725.6	705.3
46.	Wall thermocouple[e] in steam line before throttle, F	713.4	713.7	711.7	721.6	728.3	717.9	734.6	737.7	707.8	722.5	702.3
47.	Steam pressure in main line before throttle, lb/sq in. abs	411.6	409.0	410.9	407.9	408.8	413.4	414.0	416.4	418.0	419.2	407.5
48.	Steam pressure in main line after governor valve, lb/sq in. abs	397.7	395.8	395.9	390.6	399.1	351.2	353.6	262.8	185.7	186.6	394.3
49.	Steam pressure in overload line after overload valve, lb/sq in. abs.	245.0	243.9	246.4	244.7	246.7	189.3	190.7	142.9	100.5	101.1	235.3
50.	Enthalpy of steam before throttle, Btu/lb	1372.8	1371.8	1370.1	1374.4	1382.3	1373.6	1383.7	1384.4	1368.6	1375.7	1364.9
51.	Enthalpy plus velocity energy of steam leaving 21st wheel, Btu/lb	969.9	967.9	966.3	967.9	973.7	962.4	968.0	965.8	961.3	965.2	955.7
52.	Prandtl tube attack angle assuming 25° exit-flow radial angle, °	32.9	32.9	32.6	31.9	28.8	32.4	28.6	29.0	28.0	25.4	29.0
53.	Prandtl tube total press. corr'd for attack angle[10], in. Hg at 32F	1.205	1.212	1.202	1.142	1.040	0.931	0.825	0.623	0.445	0.462	0.982
54.	Prandtl tube static press. corr'd for attack angle[10], in. Hg at 32F	0.546	0.552	0.558	0.574	0.663	0.443	0.529	0.385	0.293	0.348	0.611
55.	Steam vel head leaving 21st wheel at Prandtl tube, in. Hg at 32F	0.659	0.660	0.644	0.568	0.377	0.488	0.296	0.238	0.152	0.114	0.371
56.	Steam velocity leaving 21st wheel at Prandtl tube, ft/sec	1583	1580	1558	1470	1175	1524	1160	1200	1106	905	1206
57.	Velocity energy of wet steam at Prandtl tube, Btu/lb	42.51	42.26	41.04	36.74	23.89	39.40	23.30	24.95	21.25	14.34	24.71
58.	Wet steam enthalpy at Prandtl tube, Btu/lb	927.3	925.6	925.2	931.1	949.8	923.0	944.7	940.8	940.0	950.9	931.0
59.	Energy available to turbine assuming no exhaust vel loss, Btu/lb	522.7	521.4	520.3	520.7	519.4	517.4	515.7	514.2	499.2	495.2	512.5
60.	Heat energy of steam converted to kinetic energy by turbine, Btu/lb	454.0	454.8	453.7	452.4	441.0	450.6	439.0	443.6	428.6	424.8	441.9
61.	Turbine efficiency of conversion (considered as a nozzle), per cent	86.9	87.2	87.2	86.9	84.9	87.1	85.1	86.3	85.9	85.8	86.2
	Principal Condenser Results											
62.	Moisture content of wet steam at Prandtl tube, per cent	15.1	15.3	15.3	14.8	13.3	15.2	13.4	13.3	13.0	12.2	14.9
63.	Flow by weight of dry steam to condenser, lb/hr	319000	318100	319700	318700	326000	255000	261100	195600	142500	143800	322200
64.	Flow by volume of dry steam at condenser neck, cu ft/sec	139500	135500	129500	121000	93800	126400	91900	94600	87400	71600	99100
65.	Steam velocity at condenser neck from volume flow, ft/sec	558	542	518	484	375	506	367	378	350	286	396
66.	Steam vel head at condenser neck from volume flow, in. Hg at 32F	0.0453	0.0438	0.0419	0.0389	0.0299	0.0326	0.0239	0.0184	0.0124	0.0102	0.0318
67.	Enthalpy drop of wet steam in condensing, Btu/lb	941.4	938.9	936.3	937.0	937.5	938.9	937.9	944.9	947.8	946.9	920.4
68.	Heat given up by sealing water and condensed steam, millions Btu/hr	354.2	353.0	353.9	351.0	353.0	282.8	283.2	213.5	155.8	155.7	349.1
69.	Heat loading ref'd to internal surface of 45,670 sq ft, Btu/hr sq ft	7756	7729	7749	7685	7729	6193	6200	4675	3411	3409	7644
70.	Circulating-water enthalpy rise weighted by vel traverse[f], Btu/lb	6.31	6.85	7.67	8.82	14.82	4.97	11.82	6.39	2.84	6.71	6.85
71.	Circulating-water flow through condenser by heat balance, gal/min	112100	102900	92200	79500	47560	113700	47840	66740	109600	46330	101800
72.	Circulating-water velocity through condenser tubes, ft/sec	6.91	6.35	5.68	4.90	2.93	7.01	2.95	4.12	6.76	2.86	6.27
73.	Circulating-water log. mean temperature through condenser tubes, F	36.3	36.7	37.3	38.1	41.0	35.8	39.5	36.6	34.2	37.1	45.7
74.	Circulating-water mean Reynolds number through condenser tubes	29210	27000	24460	21420	13480	29350	13220	17470	27460	12260	31180
	Ratios of log. mean temperature differences to internal heat loading:											
75.	from condenser-neck thermometers 10 inches in, F hr sq ft/1000 Btu	2.060	2.113	2.212	2.374	3.001	2.081	3.076	2.743	2.498	3.356	2.209
76.	from hotwell-condensate discharge thermocouple, F hr sq ft/1000 Btu	3.116	3.161	3.183	3.240	3.528	3.184	3.643	3.502	3.294	3.863	2.830

[a] Runs 1-64 in 1935, runs 65-88 in 1936; [b] Wt of 1 std cu ft = 0.0808 lb; [c] condensation in gage line; [d] uncalibrated but guaranteed by manufacturer to ± 0.7F at 700F, operating instrument; [e] calibrated; [f] by pitometers at location of circulating-water outlet thermocouples.

[10] Refers to footnote no. 14 of the paper

summer and winter cooling-water conditions and generator loading. Thus, in winter time, for the highest generator loading and the highest speed of the circulators, listed as test run No. 65 in Table 1, the absolute exit velocity was 1583 fps. The sound velocity was 1375 fps, and the corresponding Mach number, that is, the ratio of actual absolute velocity to acoustic velocity at that pressure, was 1.151. For the lowest loading and the slowest speed on one circulator in summer, listed as test run No. 64 in Table 1, the exit velocity was 548 fps, the acoustic velocity was 1412 fps, and the Mach number was 0.388.

The steam, after leaving the last wheel of the turbine, has to flow through an exhaust elbow before entering the condenser. The net annular flow area of the last blades of the steam turbine is 81.7 sq ft. The net area at the top flange of the condenser is 250 sq ft; the steam velocity would be reduced to nearly one third, assuming equal velocity over this last area and the same absolute pressure as at the turbine wheel. That this velocity will be equal over this area is naturally not true, but that the steam-flow distribution over this area changes systematically with changes of load, water-inlet temperature, and circulator speed is true. To prove this, the steam temperature was measured with 10-in. stem immersion mercury thermometers, at the north and south ends of the condenser neck at its center line. For the 50,000-kw runs, the steam temperature at the south end of the condenser neck is 0.4 F lower in summer time and 1.9 F higher in winter time than its companion at the north end of the condenser

DATA AND RESULTS

3[s]	15[s]	23	5[s]	9[s]	19[s]	26	31	36	29[s]	35[s]	38[s]	44[s]	42	47[s]	49[v]	52	55	53[s]	64
3/26	4/2	4/4	3/27	3/29	4/3	4/30	5/1	5/3	5/1	5/2	6/4	6/5	6/5	6/6	8/13	8/13	8/14	8/14	8/16
42.37	41.50	41.00	42.51	41.15	40.67	50.16	48.33	46.22	48.74	45.92	62.88	61.78	61.63	61.23	75.99	75.47	74.93	74.76	76.77
50360	48020	49350	40960	20760	20800	49730	41330	41030	29820	19940	50690	40780	30660	20500	49630	49290	40210	30500	19900
2	West	East	2	2	West	2	2	West	2	2	2	2	2	2	2	2	2	2	West
199.6	178.9	178.0	177.6	220.5	179.1	201.3	237.3	176.2	179.2	180.1	221.4	240.7	178.0	202.5	235.0	178.0	178.5	235.3	178.5
367	138	129	262	491	138	370	619	134	267	270	505	649	263	384	586	254	255	585	135
99.0	98.7	98.2	98.5	99.4	99.0	102.4	102.7	102.2	103.1	103.9	102.0	102.0	102.0	102.2	102.2	101.5	101.5	101.2	101.3
1.012	0.914	0.939	1.009	1.653	0.817	0.883	1.348	21.80	1.880	1.210	1.130	0.872	0.807	0.702	1.456	1.383	1.118	1.206	1.241
0.888	0.997	1.017	0.738	0.465	0.519	1.040	0.805	1.159	0.704	0.508	1.300	1.043	0.961	0.794	1.716	1.818	1.545	1.275	1.334
0.579	0.755	0.768	0.518	0.370	0.438	0.781	0.599	0.955	0.583	0.453	1.129	0.932	0.888	0.735	1.578	1.677	1.466	1.217	1.352
0.634	0.803	0.815	0.575	0.388	0.448	0.835	0.644	0.993	0.602	0.463	1.150	0.945	0.903	0.761	1.624	1.743	1.505	1.261	1.397
0.717	0.862	0.854	0.625	0.408	0.468	0.882	0.692	--[c]	0.634	0.482	1.201	0.982	0.930	0.773	--[c]	1.748	1.523	1.268	1.384
0.597	0.771	0.775	0.553	0.373	0.437	0.786	0.603	0.971	0.581	0.451	1.121	0.923	0.891	0.751	1.598	1.719	1.487	1.251	1.381
0.562	0.731	0.733	0.533	0.368	0.423	0.758	0.587	0.959	0.565	0.441	1.093	0.903	0.876	0.742	1.574	1.691	1.469	1.241	1.379
..	0.678	0.686	..	..	0.409	0.709	0.557	0.927	0.548	0.433	1.038	0.869	0.860	0.737	1.479	1.610	1.414	1.197	1.374
0.490	0.643	0.652	0.476	0.367	0.399	0.674	0.528	0.958	0.531	0.420	1.012	0.843	0.837	0.721	1.452	1.575	1.384	1.182	1.349
42.37	..	41.00	42.51	41.16	..	50.16	48.34	46.22	48.75	45.93	62.88	61.78	61.64	61.25	75.99	75.48	74.93	74.77	..
42.36	41.50	..	42.50	41.14	40.67	50.15	48.31	46.21	48.73	45.91	62.87	61.77	61.61	61.21	75.98	75.45	74.92	74.75	76.77
50.25	55.75	55.98	49.84	44.58	47.52	58.48	53.67	61.12	54.42	49.81	70.10	66.96	67.37	64.78	82.64	84.54	82.17	79.08	83.26
49.66	55.08	54.97	49.23	43.96	47.04	58.05	53.29	58.65	54.00	49.55	69.68	66.68	67.10	64.60	82.53	84.41	82.08	79.01	83.51
49.96	55.41	55.48	49.53	44.27	47.28	58.27	53.48	59.89	54.21	49.68	69.89	66.82	67.23	64.69	82.59	84.47	82.12	79.05	83.39
68.83	74.23	74.68	65.02	53.03	56.81	75.34	67.84	80.17	65.08	57.67	84.35	78.49	76.55	71.16	94.69	96.87	92.08	86.32	89.40
68.43	73.92	74.28	64.60	52.93	56.72	74.86	67.47	79.94	64.92	57.57	84.15	78.32	76.58	71.18	94.65	96.91	92.10	86.38	89.39
62.31	67.76	67.51	61.25	56.95	58.77	68.03	61.37	61.16	59.39	56.31	78.17	73.09	72.87	68.89	89.22	92.10	87.88	82.62	86.98
78.50	80.74	80.61	77.83	77.94	79.39	79.36	77.73	79.23	74.65	75.92	85.69	81.22	79.94	76.37	89.33	92.22	89.22	84.00	88.26
64.28	71.73	72.11	61.78	..	55.54	72.61	65.07	78.39	63.78	56.29	82.84	76.90	75.69	70.48	93.73	96.41	91.73	86.09	89.18
63.80	71.84	71.86	61.41	..	55.69	72.54	64.83	78.89	63.77	56.65	83.20	77.46	76.04	70.98	94.10	96.58	92.06	86.48	89.45
68.48	73.86	74.26	64.74	52.98	56.67	74.85	67.39	79.77	64.97	57.57	84.21	78.27	76.56	71.15	94.61	96.98	92.10	86.43	89.45
81.2	81.1	80.3	80.5	77.6	77.0	79.5	76.4	84.6	74.0	77.8	90.0	84.6	83.8	79.7	87.0	90.4	94.4	89.6	93.1
94.1	92.4	92.2	93.8	88.9	93.8	88.2	87.8	93.7	88.2	92.4	97.8	96.2	96.8	95.5	93.8	96.0	102.8	97.6	100.6
74.0	76.2	75.7	73.5	70.0	71.3	75.1	71.2	76.1	70.2	68.8	83.4	78.6	77.7	74.9	83.0	86.4	88.0	83.4	86.4
80.7	78.7	76.5	80.4	78.0	77.2	76.7	73.4	78.2	72.6	73.2	84.2	78.2	77.8	76.1	81.1	84.5	87.1	85.1	85.5
82.1	83.1	82.4	82.7	82.5	83.1	82.6	81.1	82.3	79.9	79.4	89.3	81.3	81.7	80.4	85.7	89.6	93.2	88.7	92.4
78.7	75.0	73.9	77.4	79.7	80.7	76.8	77.6	77.8	76.6	77.5	76.2	77.3	80.9	80.7	75.8	74.6	76.4	78.8	79.7
1760	1760	1790	1670	1510	1510	1770	1680	1670	1580	1510	1780	1670	1570	1510	1780	1790	1670	1580	1500
52120	49780	51140	42630	22270	22310	51500	43010	42700	31400	21450	52470	42450	32230	22010	51410	51080	41880	32080	21400
6090	5920	2810	5430	2390	2410	5570	5510	0	3660	2220	5300	5140	3550	2200	4710	4650	4840	3410	1980
46030	43860	45330	37200	19880	19900	45930	37500	42700	27740	19230	47170	37310	28680	19810	46700	46430	37040	28670	19420
497100	474700	491200	396100	201300	209800	494800	396500	363400[w]	289500	197900	506800	397000	301000	209300	512300	512200	413500	309700	218600
74100	41400	65900	0	0	0	83800	0	0	0	0	99400	0	0.	0	119700	119900	0	0	0
3410	4180	4550	4170	2180	2320	4730	4030	3600	2960	2290	4920	4340	3450	2520	4530	4390	4280	3400	2770
115300	105900	109800	85800	36800	38800	109800	85200	0	56900	35400	109500	81400	57000	35900	106500	106200	81400	56500	34300
378500	364300	376800	306100	162300	168600	380300	307400	359800	229700	160200	392300	311300	240600	170900	401200	401600	327900	249700	181500
414.9	410.7	410.4	414.6	417.8	402.7	412.0	416.2	404.8	412.0	409.5	410.2	408.9	406.6	395.4	397.1	394.4	385.4	391.7	365.1
706.0	704.3	707.5	707.3	744.7	701.4	712.1	737.7	745.5	738.1	735.6	718.8	732.5	740.0	736.1	727.7	731.0	713.9	739.1	736.2
702.4	701.5	703.8	705.0	743.3	696.5	710.4	728.4	740.7	734.2	731.5	716.2	729.3	736.0	731.0	724.8	727.8	712.2	735.4	730.4
405.4	406.8	402.4	410.3	415.4	416.1	404.0	409.1	410.0	412.5	418.1	405.6	409.7	412.3	419.3	400.1	399.3	403.5	409.3	408.6
395.3	397.2	398.4	352.6	183.2	184.6	395.2	359.2	333.1	262.1	179.4	397.2	359.8	274.6	189.2	391.9	392.1	367.4	280.4	195.9
235.7	229.4	239.4	186.8	101.1	100.4	247.4	195.0	183.1	144.3	99.1	253.0	193.0	147.7	103.3	258.3	258.2	198.5	152.6	107.2
1365.4	1364.4	1366.4	1365.9	1386.4	1362.2	1369.0	1382.9	1387.2	1383.0	1381.4	1372.5	1380.0	1384.1	1381.6	1377.8	1379.8	1370.0	1383.8	1382.1
958.4	958.5	963.0	951.3	968.6	959.5	965.2	966.7	982.4	971.0	971.9	971.5	971.1	977.5	986.2	990.9	995.6	984.6	992.1	1017.0
28.9	26.0	26.0	27.5	25.0	25.6	26.2	26.3	25.0	25.1	30.2	26.6	29.7	35.2	36.0	34.0	34.4	40.9	42.5	59.0
1.014	1.063	1.084	0.813	0.437	0.545	1.112	0.863	1.206	0.752	0.534	1.351	1.095	1.019	0.847	1.815	1.923	1.668	1.381	1.491
0.637	0.790	0.804	0.554	0.383	0.449	0.819	0.629	0.982	0.599	0.465	1.154	0.954	0.910	0.754	1.616	1.717	1.502	1.248	1.380
0.377	0.273	0.280	0.259	0.104	0.091	0.293	0.234	0.224	0.133	0.071	0.197	0.140	0.109	0.093	0.199	0.206	0.166	0.133	0.111
1195	943	354	1072	836	732	967	373	793	774	633	689	637	591	583	598	594	561	556	548
24.37	15.40	15.65	19.58	12.30	9.34	16.12	16.39	11.09	10.53	7.08	8.28	7.10	6.15	6.06	6.34	6.28	5.55	5.52	5.40
934.0	943.1	947.3	931.7	956.3	950.2	949.1	950.3	971.3	960.5	964.8	963.2	964.0	971.3	980.1	984.6	989.3	979.1	986.6	1011.6
510.9	499.7	500.8	503.7	495.3	475.4	501.4	508.0	485.3	492.7	482.8	487.5	487.2	476.4	462.6	473.7	472.0	460.6	461.8	435.7
439.3	426.1	426.1	434.2	430.1	412.0	428.1	432.3	415.9	422.5	416.6	418.5	416.0	412.8	401.5	403.4	400.7	391.0	397.2	370.5
86.0	85.3	85.1	86.2	86.8	86.7	85.4	85.2	85.7	85.8	86.3	85.9	85.4	86.7	86.8	85.2	84.9	84.9	86.0	85.0
14.7	14.1	13.8	14.7	11.9	12.7	13.6	13.1	11.8	12.1	11.3	12.8	12.4	11.7	10.5	11.3	10.9	11.7	10.7	8.40
323000	313000	325000	261200	143000	147300	328500	267100	317500	202000	142100	342200	272600	212500	152900	356100	357800	289600	223100	166300
95400	71300	73500	83100	..	57400	73000	76000	58300	59700	53700	55000	52400	42600	36000	41300	38500	35700	32400	22050
382	285	294	332	..	230	292	304	233	239	215	220	209	170	144	165	154	143	130	88
0.0306	0.0221	0.0236	0.0215	..	0.0083	0.0237	0.0201	0.0182	0.0119	0.0075	0.0185	0.0140	0.0089	0.0054	0.0144	0.0135	0.0101	0.0071	0.0036
922.0	916.6	920.7	918.7	947.7	934.8	922.3	931.2	934.5	938.1	946.3	919.3	924.8	932.9	947.0	928.2	930.7	924.5	937.7	959.6
349.5	334.4	347.4	281.7	154.2	158.0	351.3	286.8	341.1	215.9	152.0	361.2	288.4	224.9	162.3	372.9	374.3	303.6	234.6	174.6
7654	7322	7607	6168	3386	3459	7691	6281	7468	4727	3328	7909	6313	4924	3553	8165	8194	6648	5137	3821
7.61	13.95	14.50	7.05	3.15	6.65	8.10	5.15	13.74	5.47	3.76	6.99	5.03	5.58	3.45	6.59	8.98	7.17	4.28	6.61
91700	47880	47850	79800	97700	47440	86600	111200	49570	78800	80700	103200	114500	80500	93900	113000	83300	84600	109500	52700
5.65	2.95	2.95	4.92	6.02	2.93	5.34	6.86	3.06	4.86	4.98	6.37	7.06	4.96	5.79	6.97	5.13	5.21	6.75	3.25
46.4	49.2	49.0	46.3	42.8	44.2	54.5	51.1	53.7	51.7	47.9	66.6	64.5	64.7	63.1	79.5	80.4	78.9	77.1	80.5
28450	15540	15490	24700	28510	14220	30480	37200	17260	26600	25660	43240	46730	32800	37460	56100	41800	41600	52700	26480
2.308	3.091	3.022	2.483	..	3.291	2.349	2.212	3.344	2.559	2.562	2.072	2.014	2.276	2.150	1.762	1.967	1.963	1.792	2.312
2.895	3.387	3.330	2.974	2.997	3.600	2.648	2.614	3.520	2.802	2.898	2.218	2.197	2.424	2.282	1.852	2.018	1.993	1.811	2.329

[s]Fluctuations during run of back pressure at south wall of condenser neck less than 0.005 in. Hg; unmarked runs, about 0.010 in. Hg; [v]about 0.030 in. Hg; [w]calculated from weigh-tank measurements of hotwell condensate.

neck, and this change takes place gradually from summer to winter. This is shown in Fig. 3. The explanation of this phenomenon can be found in the velocity direction of the steam leaving the last wheel of the turbine. Under summer conditions these velocities are small and directed with the wheel rotation while in winter they are large and opposite to this rotation.

On the other hand, the steam temperature, measured 10 in. in at the south end of the condenser neck, was lower than the steam-saturation temperature corresponding to the pressure measured directly at the wall. At 50,000-kw load this difference, which was as low as 0.2 F in summer, had a maximum value of 1.6 F in winter. This excess, which gradually increases with decreasing hotwell-condensate temperature as shown in Fig. 4, is a velocity-head effect. Obviously, the pressure measured at the condenser-neck wall is not a static pressure, but includes a part of the velocity head of the steam.

The velocity energy of the steam at the condenser neck can be recovered in the hotwell by converting it into pressure. Because of the large center lane and the east outer lane formed by the shell and the tube bundle, the steam is compressed at the bottom of the condenser toward the hotwell where it has to come to a stop. To prove this, the average velocity head of the steam at the condenser neck has been calculated from its volume based on the two mercury-thermometer readings. The measured pressure increase from the condenser neck to the hotwell is shown as a function of this velocity head in Fig. 5. It is therefore very important for reheating the condensate, that there are open lanes which admit the steam directly to the hotwell, especially if the absolute pressures are very low and the velocities are high. The steam leaving the last wheel has a very high velocity component along the axis of the turbine, which will tend to compress the steam more at the far end (in this case, at the east or generator

end) than at the near end, and it is for that reason that the lane for admission of the steam to the hotwell should be at the far end. Fig. 5 shows that the rise of steam pressure from the condenser-neck wall to the hotwell is nearly three times the average velocity head at the condenser neck. That more than this assumed theoretical velocity head is recovered is due (1) to the velocity energy already in the steam when entering the condenser

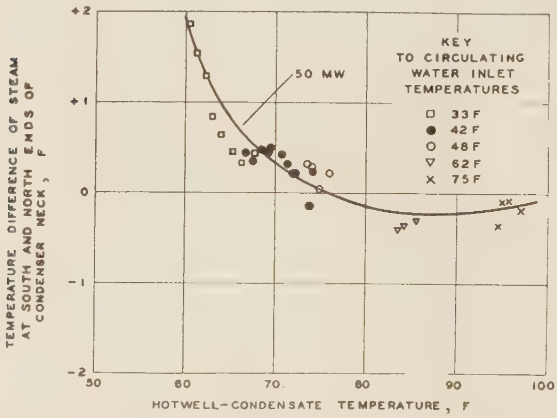

Fig. 3 The Systematic Change Between the North and South Steam Temperatures in the Condenser Neck From Winter to Summer Conditions at 50,000-Kw Generator Loads

(Since the steam is wet, the temperature difference represented by the ordinates corresponds to a pressure difference. This is a flow effect caused by the reversal in the rotation of the steam leaving the last wheel of the turbine from summer to winter conditions. Any ordinate, which is item 27 minus item 28 of Table 1, is the difference between the temperatures measured by the direct-exposed mercury thermometers with their bulbs 10 in. in from the inside surface of the condenser-neck wall. The abscissas are items 24 of Table 1; any condenser steam temperature or pressure can be used instead.)

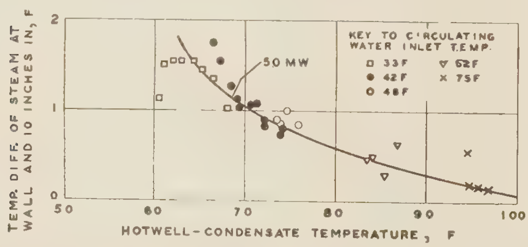

Fig. 4 The Effect of Decreasing Condensate Temperature on the Steam Pressure at the South Wall of the Condenser Neck at 50,000-Kw Generator Loads

(Since the steam is wet, the temperature difference represented by the ordinates corresponds to a pressure difference. This difference is caused by the horizontal flange at the bottom of the south wall of the condenser neck, 19 in. below the pressure connection. This projects about 5 in. in so that the measurement 10 in. in is influenced by it only to a minor degree, while the measurement taken at the wall surface includes a part of the velocity head of the steam flow. At a constant generator load, the velocity head increases with decreasing condensate temperature, because of the increase in the specific volume of the steam. Any ordinate is the steam saturation temperature corresponding to item 12 minus item 27 of Table 1. The abscissas are listed as item 24.)

neck, and (2) to the higher velocities at the far end of the condenser neck.

Since the pressure measured at the condenser-neck wall is lower than that in the hotwell, it is clear that the pressure measured at the condenser-neck wall is not the total pressure at this point. It has already been shown that it is not the static pressure. It is apparent, therefore, that the pressure measured at the condenser-neck wall by the usual means is erroneous because it represents neither the total nor the static pressure, but something between them—all dependent on the flow.

The Existence of Condensate Superheating. Since the steam is wet, a rise of pressure from the condenser-neck wall to the hotwell corresponds to a rise of steam temperature. On the basis of the interpretation of this rise as a natural flow effect, the author claims to have the solution of a question of much dispute; that is, the superheating of the hotwell condensate. This is discussed in detail in the following.

Undercooling or superheating of the condensate is understood as the temperature difference between the steam entering the condenser at its neck and the condensate in the hotwell. In customary practice the temperature of the steam at the condenser neck is taken as the saturation temperature corresponding to the absolute pressure at the wall. When this steam temperature minus the condensate temperature is positive, there is undercooling and when negative, superheating. The curve in Fig. 6 shows the superheating of the condensate based upon the absolute pressure at the south wall of the condenser neck. That this phenomenon of superheating the condensate has been

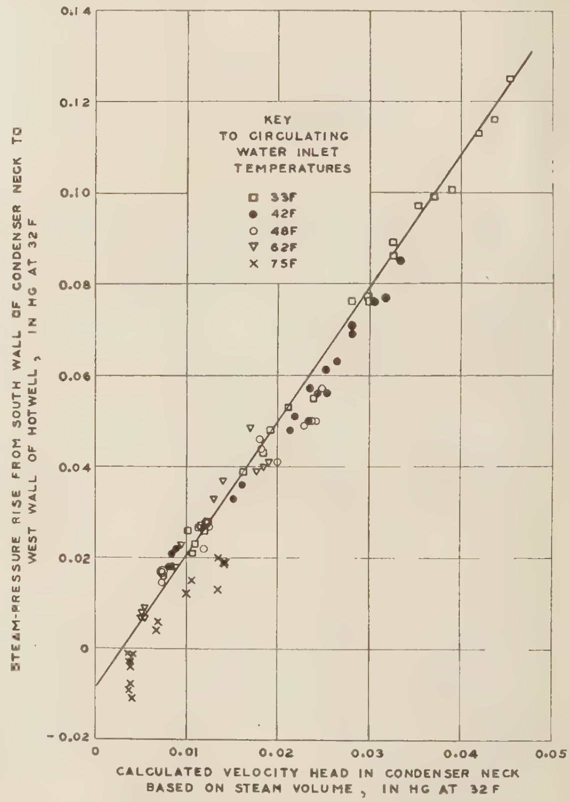

Fig. 5 Recovery in the Hotwell of the Velocity Head of the Steam at the Condenser Neck

(The steam-pressure rise represented by the ordinates is the measured difference between the saturation pressure corresponding to the temperature indicated by the thermocouple exposed to the steam at the west wall of the hotwell (item 23 of Table 1) and the pressure measured by the absolute-pressure oil manometer connected to the hole in the south wall of the condenser neck (item 12 of Table 1). The calculated velocity head represented by the abscissas (item 66 of Table 1) is obtained from the wet-steam flow rate to the condenser (item 43 of Table 1), the moisture content of the steam (item 62), the specific volume of the steam at the average north and south condenser-neck steam temperatures (items 27 and 28), and a net flow area of 250 sq ft for the condenser neck. Eighty-four test points are shown.)

doubted[3] and often contradicted is due to the fact that this does not take place in all condensers and depends upon (*a*) the design of the condenser, (*b*) the conditions under which it is operated, and (*c*) the point where one measures the condenser-neck steam pressure or temperature and what kind of pressure (temperature) is measured at the point selected, that is, total

[3] "The Surface Condenser," by B. W. Pendred, Isaac Pitman & Sons, London, 1935, p. 136.

or static. These three conditions are discussed in the following three paragraphs.

(*a*) The old practice has been a shell filled solidly with tubes which "killed" the velocity head of the steam at the top tubes. Later designs, such as the eccentric-shell type, the radial-flow type, and the type with two banks, that is, one on each side of the shell, or any combination thereof, provide a direct admission of the steam to the hotwell and make the condensate superheating phenomenon possible.

(*b*) This phenomenon of superheating is most evident in winter time, with low condensate temperatures, high vacua, and with high loads, as shown in Fig. 6. The highest superheating occurred during these tests at the high load and high circulating-pump speeds for the lowest inlet-water temperature, that is, it was 6.1 F for run No. 65 in which these conditions were 50,000-kw load, both circulators at 240 rpm pumping 112,000 gpm of

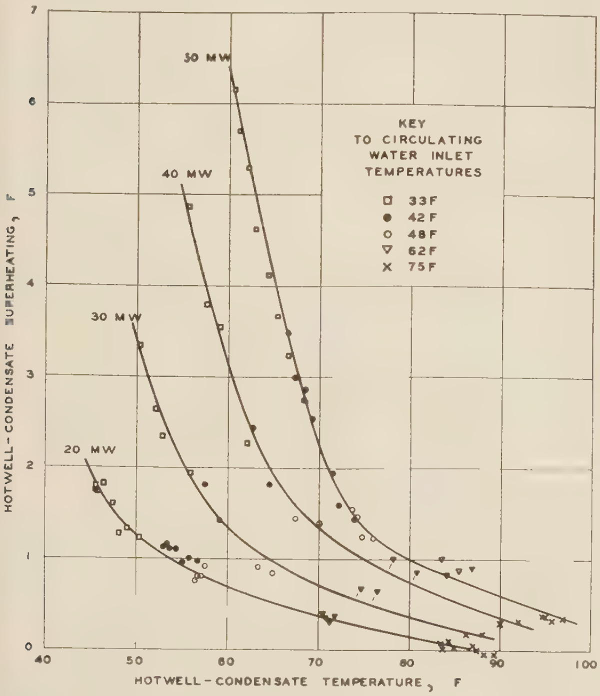

Fig. 6 Superheating of the Hotwell Condensate at Various Condensate Temperatures for Different Turbogenerator Loads

(The designation of each curve is the generator load in megawatts. Since the steam is wet, the temperature difference represented by the ordinates corresponds to a pressure difference. This graph can be regarded as a representation in different form of the relation shown in Fig. 5, because the velocity head of the steam at a constant load increases with a decrease of condensate temperature (which corresponds to a decrease of pressure) owing to the increase in specific volume of the steam and consequently its velocity; and because at a given condensate temperature (steam pressure) the velocity head of the steam increases quite rapidly with an increase of generator load. In this connection it may be pointed out that the generator load corresponds roughly to a definite steam flow by weight to the condenser for any condition of condensate temperature (steam pressure). Each ordinate is the elevation of the hotwell-condensate temperature (item 24 of Table 1) above the saturated-steam temperature corresponding to the pressure at the south wall of the condenser neck (item 12). The abscissas are listed as item 24 of Table 1.)

water at 33.0 F inlet temperature. In the case of this condenser, the condensate superheating was less than 2 F for (1) all load conditions when the condensate temperature was above 70 F, the equivalent of a hotwell pressure of 0.75 in. Hg; or for (2) the low-load runs of 20,000 kw, even when the condensate temperature was as low as 45 F, the equivalent of 0.30 in. Hg abs, as can be seen from Fig. 6.

(*c*) As already mentioned, undercooling is customarily assumed to be determined by the temperature difference of the steam in the condenser neck and the condensate in the hotwell. The temperature of flowing wet steam is determined by its static pressure, which varies widely at different points in the condenser neck because of the nonuniformity of the flow of the steam. The magnitude of the temperature variations due to this flow condition is of no slight degree, as shown in Table 4.

Consider run No. 65, for which the highest steam velocities were obtained. If the shielded-thermocouple temperature

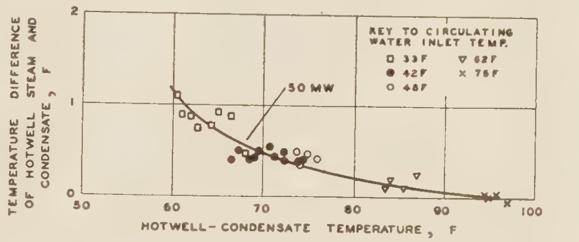

Fig. 7 Increase of the Hotwell-Condensate Depression With the Decrease of the Hotwell-Condensate Temperature for 50,000-Kw Generator Loads

(Since the steam is wet, the temperature difference represented by the ordinates corresponds to a pressure difference. This difference is a minor velocity-head effect of the steam flow similar to the one shown in Fig. 5, but presented in the variables used in Fig. 6. The ordinates are the differences between item 23 and item 24 of Table 1. The abscissas are listed in Table 1 as item 24.)

measurements can be relied upon, the steam temperature in the condenser neck showed a variation of 4.1 F. The average of the six shielded thermocouples, compared with the average of the two 9-in. immersion mercury thermometers was higher by 0.9 F; and compared with the saturation temperature corresponding to the oil-gage pressure at the south wall this thermocouple average was lower by 1.2 F.

For instance, if the superheating is calculated from the steam temperature at the north end of the condenser neck, measured by the mercury thermometer reaching 9 in. in, one finds it to be 9 F instead of the 6.1 F as calculated from the absolute pressure at the south wall of the condenser neck. It is clear that condensate undercooling, or superheating, is highly dependent on the choice of the condenser-neck steam temperature.

If undercooling is referred to condenser-neck steam temperature (static pressure), it has no absolute significance because considerable variations of static pressure, and consequently of steam temperature, will always exist due to the nonuniformity of the steam flow in the condenser neck. This conclusion obviously applies to all condensers, since the condenser-neck steam flow is never uniform for any condenser.

The author suggests that the loss which undercooling is supposed to evaluate should not be referred to the steam temperature (static pressure) at any arbitrarily chosen point or average of points in the condenser neck, but that it should be based on the steam saturation temperature corresponding to the total pressure, if this were known. With this procedure, superheating would never be encountered, since the total pressure of a flowing fluid never rises along the path of flow. For example, the temperature of the hotwell condensate can never be higher than the temperature of the steam in the hotwell, since the steam pressure measured there is practically a total pressure. In fact, a minor depression of the condensate temperature generally exists, which for this condenser, as shown in Fig. 7, had a maximum value of only 1.1 F and is related to the velocity head of the steam.

The condenser-neck wall pressure of this condenser, which previously, in Fig. 4, has been proved to be not truly a static pressure because of the influence of the flange below it, is ob-

viously not a total pressure, because it is so much lower than the hotwell pressure. It is neither a static nor a total pressure; but it has been made evident in the foregoing that even if it were truly the static pressure, it would be an arbitary measurement governed by the local steam velocity.

Summary. With the old type of condenser, in which the flow loss from the condenser neck to the hotwell is high, the undercooling of the condensate as originally defined was made up principally of a condenser pressure drop. In the modern design of a condenser which is provided with the proper type of well-located open lanes, undercooling has, however, reversed itself, because it shows principally the degree of recovery by the hotwell of the velocity at the condenser neck.

It is not surprising that the measurement of the superheating of the condensate has been a doubtful one, but the test data prove that if it is referred to static pressure taken at the condenser neck it can really exist, and that it increases with the velocity head of the steam at the condenser neck. For any condenser which exhibits superheating, its magnitude will be different, depending on the design of the condenser, but will still depend upon the conditions of operation which affect the velocity head of the steam in the condenser neck.

Hotwell condensate undercooling, if it is referred to the total pressure at some initial point arbitrarily selected in the path of the steam flow ahead of the hotwell, will be a quantity which is arbitrary only to the extent that it depends upon the particular friction loss of the flow between the two points. But it must be referred to temperatures corresponding to total pressures at the initial point; because the undercooling referred to the condenser-neck wall pressure, or any static pressure or any temperature as ordinarily measured (static pressure), is unnecessarily arbitrary since it neglects the temperature recoverable from the steam velocity.

(*B*1) The Resistance to Heat Flow

As a basis of comparison of different test runs, the author introduces the unit of *heat loading H* in Btu per hr sq ft, and defines it as the heat transferred per unit time and unit of surface.

The general formula used in heat transfer is

$$UAt = q \qquad [1a]$$

This equation can be written as

$$At = qr \qquad [1b]$$

or

$$r = t/H \qquad [1c]$$

In this paper, Equation [1*c*], will be used.

Nomenclature[4]

- U = overall heat-transfer coefficient, Btu per hr sq ft F
- A = cooling surface, sq ft
- t = some temperature difference between the steam and cooling water, F
- q = heat transferred per unit time, Btu per hr
- r = overall thermal surface resistivity to heat flow = $1/U$, F hr sq ft per Btu
- H = heat loading = q/A, Btu per hr sq ft

In customary condenser practice, A is the outer cooling surface of the tubes between the tube heads; q is the quantity of heat necessarily removed from the steam to condense it, and is determined by subtracting the enthalpy of the condensate from the enthalpy of the steam leaving the turbine. The temperature difference t is taken as the logarithmic mean temperature difference based on the inlet and outlet temperature of the cooling water and the saturation temperature corresponding to the steam pressure measured at the condenser-neck wall. The coefficient U is as defined by Equation [1*a*]. It has been the author's practice to use the reciprocal of the heat-transfer coefficient $1/U$ which is equal to the resistivity because of the advantage that its component parts are directly additive. The resistivity times the heat loading gives the temperature difference, or $rH = t$.

In common condenser practice, this heat loading (referred to external surface) is from 2000 to 12,000 Btu per hr sq ft. The maximum external heat loading on this condenser was only 7500 Btu per hr sq ft because it was not originally designed for chlorination of the circulating water and therefore was amply proportioned. During all test runs, chlorination was used and for this reason any changes in the resistances due to changes in the dirtiness of the tubes have been neglected.

The resistivity r is affected by so many independent variables such as circulating-water temperature, velocity, and steam temperature, that any analysis of its behavior appears to be a complex task. It develops, however, that a theoretical study indicates that r is affected by the heat loading to a minor degree only, and that its dependence upon the water temperature and velocity can be incorporated into a single criterion of similarity, the Reynolds number.

Theoretical. No matter how well a condenser performs, it will never have a resistivity lower than that which can be attained by the single tube. The purpose of this theoretical presentation is to give an easy grasp of the tendency of the resistance of heat flow in a single tube even though its absolute value is not fully and exactly established.

The resistivity r, can be split into r_i, the resistivity to heat flow of the cooling-water film at the inside of the tube; r_t, the resistivity of the tube wall itself; and r_o, the resistivity of the condensate film at the outside of the tube. The Equation [1*c*] then becomes

$$r = r_i + r_t + r_o = t/H \qquad [1d]$$

The Resistivity of the Tube Wall r_t. This resistivity is very small for clean condenser tubes, or 0.062×10^{-3} F for 1 Btu per hr sq ft. It is directly computed from the thermal conductivity of brass, which is 63 Btu per hr ft F. For an abnormally high average heat loading of 16,000 Btu per hr sq ft, which occurs for 0.5 sq ft per kw, the temperature drop is only 1.0 F. Since the highest average internal heat loading of this condenser is about 8000 Btu per hr sq ft, the tube wall temperature drop is only 0.5 F. If this 0.5 F is compared with the overall temperature drop of this condenser, which is of the order of 20 F, it is evident that this resistance is negligible and averages only 1 to 3 per cent of the total resistance, which is less than the possible error of the testing.

For the basis of the calculation of the resistivity r_i (water to wall), the test results of Eagle and Ferguson[5] are used, which were obtained in the range of usual condenser conditions of heat loading, cooling-water velocity, and temperature. These results were given by Nusselt in 1931[6] in the form,

$$(\mathrm{Nu}) = 0.02114\ (\mathrm{Pr})^{0.365}\ (\mathrm{Re})^{0.819} \qquad [2]$$

where (Nu) = Nusselt number,[7] (Pr) = Prandtl number, and

[4] For symbols used in this paper see "Symbols for Heat and Thermodynamics," American Standards Association, 29 West 39th Street, New York, N. Y., February, 1931.

[5] "The Coefficient of Heat Transfer From Tube to Water," by A. Eagle and R. M. Ferguson, Proceedings of the Institution of Mechanical Engineers, vol. 2, November, 1930, pp. 985–1075.

[6] "Der Wärmeaustausch zwischen Wand und Wasser im Rohr," by W. Nusselt, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 2, 1931, pp. 309–313.

[7] "Heat Transmission," by W. H. McAdams, McGraw-Hill Book Company, New York, N. Y., 1933, p. 96.

(Re) = Reynolds number. Each of these numbers is dimensionless and requires no particular system of units of measurement, but only a consistent set of units in the calculation of the number. The resistivity r_i can be evaluated from the Nusselt number, which is the water-film heat-transfer coefficient (referred to internal surface) times the tube inside diameter divided by the thermal conductivity of the water. The Prandtl number is a function only of the temperature of the water, being the product of its specific heat and absolute viscosity, divided by its thermal conductivity. The Reynolds number is the product of the water velocity in the tube and the inside diameter of the tube divided by the kinematic viscosity of the water.

It should be noted especially that the Reynolds number is directly proportional to the water velocity at a constant water temperature, and that a change in the temperature of the water from 32 to 80 F increases the Reynolds number 100 per cent at a constant water velocity. The Reynolds number is simply a means of correlating the cooling-water temperature and velocity.

Equation [2] is somewhat modified by using the new thermal conductivity of water as determined by Schmidt and Sellschopp in 1932.[8] It then becomes

$$(Nu) = 0.02260\,(Pr)^{0.365}\,(Re)^{0.819} \qquad [3]$$

Equations [2] and [3] refer to the conditions at the outlet-water end of the tube with a ratio of length to diameter (l/d) of

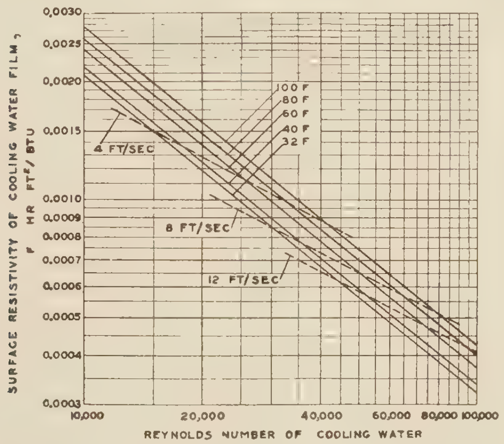

FIG. 8 MEAN SURFACE RESISTIVITY OF THE COOLING-WATER FILM ON THE INSIDE OF A 0.902-IN. INSIDE DIAMETER HORIZONTAL TUBE

(The broken curves represent constant velocity of the cooling water through the tubes. The solid curves represent constant logarithmic mean temperature of the cooling water. The surface resistivity, which is referred to the internal surface of the tube, is the mean resistivity for the heated 23.75-ft length of the tube and is calculated from Equation [4]. Equation [4] is the application to this case of the results of the tests of Eagle and Ferguson,[5] which were made on electrically heated brass condenser tubes from 0.5 to 1.5 in. outside diameter with heated lengths of about 6 ft, over a range of internal-heat loading of 4000 to 20,000 Btu per hr sq ft, a range of water velocities from 3 to 11 fps, and a range of water temperatures from 40 to 130 F. The surface resistivities of the figure have been calculated for an internal-heat loading of 7760 Btu per hr sq ft, but are applicable to heat loadings from 0 to 20,000 Btu per hr sq ft with a maximum error of only 1 per cent in the resistivity.)

approximately 94 as in the tests of Eagle and Ferguson.[5] On the basis of other tests,[6] Equation [3] receives a small correction of only 1 per cent (a) when applied to the condenser being discussed in this paper, the tubes of which have a ratio of length to diameter of 316, and (b) when corrected to the mean condition over the entire length of the tube instead of the end of the tube. Equation [3] then becomes

$$(Nu) = 0.02239\,(Pr)^{0.365}\,(Re)^{0.819} \qquad [4]$$

[8] "Wärmeleitfähigkeit des Wassers bei Temperaturen bis zu 270 C," by E. Schmidt and W. Sellschopp, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 3, 1932, pp. 277–286, Fig. 10.

Equation [4] can be applied to any condenser tube for which l/d is greater than 250, with an error of less than $2^1/_2$ per cent in (Nu) or the mean resistivity r_i. The term "mean resistivity" is used to distinguish the mean resistivity over the whole length of the tube from that at the outlet end. The changes in this resistivity due to heat loading are negligible.

Equation [4], which is cumbersome for calculations of r_i, is given in Fig. 8 in graphical form for a tube with an inside diameter of 0.902 in. and for the range of usual condenser practice. It is

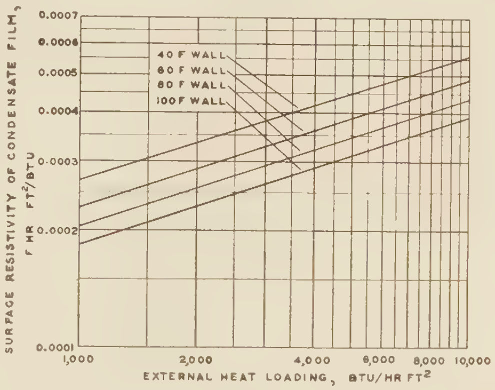

FIG. 9 THEORETICAL SURFACE RESISTIVITY OF THE CONDENSATE FILM OF QUIESCENT SATURATED STEAM ON A HORIZONTAL 1-IN. OUTSIDE DIAMETER TUBE ACCORDING TO NUSSELT[9]

(The temperatures refer to the wall temperatures on the outside surface of the tube. The resistivity represented by the ordinates is referred to the external tube surface, and is the reciprocal of α in Nusselt's formula, which in Nusselt's symbols is

$$\alpha = 0.725\sqrt[4]{\frac{r\gamma^2\lambda^3}{\eta d\,\Delta t}}$$

where α = the heat-transfer coefficient, Btu per sec sq ft F referred to the external surface of the tube; d = the outside diameter of the tube, ft; Δt = the difference between the steam temperature and the temperature t_w of the outside surface of the tube wall; and the following properties of water taken at the temperature ($t_w + {}^1/_2\,\Delta t$): r = latent heat of evaporation, Btu per lb; γ = density, lb per cu ft; λ = thermal conductivity, Btu per ft sec F; and η = absolute viscosity in force units, lb sec per sq ft.)

apparent that the resistivity is a function of the Reynolds number and the temperature of the cooling water. For a Reynolds number of 30,000 at a logarithmic mean water temperature of 62 F and a water velocity of 4.7 fps through a 1-in. tube with a wall thickness of 0.049 in., a 0.001 F temperature drop per unit heat loading is obtained; thus, r_i = 0.001 F hr sq ft per Btu, based on the internal tube surface. Thus, for an internal heat loading of 8900 Btu per hr sq ft, the temperature drop through the cooling-water film would be 8.9 F.

To compute the resistivity r_o, Nusselt's[9] formula is used, which is based on film condensation of dry steam vapor at rest with a horizontal tube and is referred to external surface. In order to dispose of this elaborate equation, the graphical method shown in Fig. 9 has been used. Here the resistivity r_o is plotted as a function of the heat loading, which is referred to the external surface, called external heat loading. It shows that the resistivity increases somewhat with an increase of the heat loading and with the lowering of the wall temperature.

The following gives some conception of the importance of the different resistivities. With the conditions of the example already given for r_i, the tube-wall temperature is calculated to be 72 F. At this wall temperature and the heat loading of the example, which is 8000 Btu per hr sq ft when referred to external surface, the resistivity of the outer condensate film is 0.42×10^{-3} F hr sq ft per Btu, according to Fig. 9. The

[9] "Die Oberflächenkondensation des Wasserdampfes," by W. Nusselt, *Zeitschrift V.D.I.*, vol. 60, 1916, p. 573. formula [69c].

total r is thus 1.4×10^{-3}, or r_o is 30 per cent of r, and r_i is 70 per cent of r.

At this point it will be remarked that the total resistivity r can be referred either to internal or external tube surface. Although the practice in the usage of the overall heat-transfer coefficient has been to refer it to the external surface, the author refers the overall r to the internal surface since it is predominantly influenced by r_i rather than r_o. Naturally, in the addition involving r_o, which is referred to external surface, it must be referred to internal surface and in the process it is increased by approximately 10 per cent in the case of this tube size.

The separate resistivities have been incorporated and plotted as a function of the Reynolds number in Fig. 10. It is evident that the effect of heat loading is assuming a secondary rôle and that at any given Reynolds number the increase of resistivity of the cooling-water film with higher circulating-water temperature

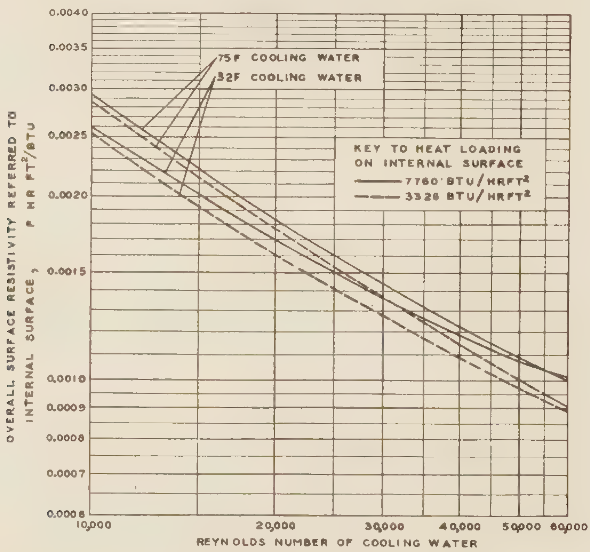

FIG. 10 OVERALL THERMAL SURFACE RESISTIVITY FROM QUIESCENT LOW-PRESSURE SATURATED STEAM TO WATER FLOWING THROUGH A SINGLE HORIZONTAL BRASS TUBE WITH AN OUTSIDE DIAMETER OF 1 IN. AND A WALL THICKNESS OF 0.049 IN.

(The resistivity is referred to internal surface. The two internal-heat loadings indicated correspond approximately to the highest and lowest average heat loadings on the condenser tested, obtained at generator loads of 50,000 and 20,000 kw. The resistivities are calculated by the addition of the resistivity of Fig. 8 and that of Fig. 9, corrected to a basis of internal-heat loading, and by taking a thermal-conductivity value for brass of 63 Btu per ft hr F.)

is virtually offset by the decrease in resistivity of the condensate film because of the higher wall temperature, giving a bundle of curves very close together.

The importance of Fig. 10 is that it shows the lowest resistivity attainable as a limit for this condenser with 1-in. tubes. It also suggests that the use of the Reynolds number is the logical procedure to adopt for comparing condenser test results at different water-inlet temperatures. The substitution of the use of the Reynolds number for the water velocity has reduced to one criterion of similarity the effects of two independent variables, that is, the cooling-water velocity and temperature.

Application to the Whole Condenser. The principal problem in the study of the resistivity r for the whole condenser is the selection of the proper temperature t to be used in the calculation of r. Many equations have been suggested for deriving the proper value of t from condenser test results[10] to be used in solving for r from the equation $r = t/H$. Most authors use a logarithmic mean temperature difference between the steam and the cooling water as the most precise one; the simplest one is:

$$t = \frac{t_2 - t_1}{\log_e \dfrac{t_c - t_1}{t_c - t_2}}$$

where t_2 = cooling water temperature leaving the condenser, t_1 = cooling water temperature entering the condenser, and t_c = steam temperature corresponding to the pressure at the condenser neck.

In customary practice this formula takes for t_c the steam-saturation temperature corresponding to the condenser-neck wall pressure and assumes that this temperature is representative for the steam in the whole condenser neck and tube bank. That this assumption is not true has already been stated in the discussion of the steam flow entering the condenser (see *A*1), where it is mentioned that during run No. 65 a temperature variation of 4 F over the area of the condenser neck has been recorded.

Actually, the steam temperature, as also stated previously under *A*1, increases at the outside of the tubes from the top to the bottom of the condenser. In the worst case, run No. 65, a steam-temperature rise of 6.1 F is not considered if the customary calculating procedure is followed. The tubes at the hotwell are exposed to 24.1 F temperature difference instead of 18 F as at the condenser neck, an increase of 34 per cent in temperature difference. The tubes at the hotwell will therefore transmit more heat, and assuming that one half of the tubes is exposed to this higher temperature difference, and assuming the same mean heat loading, it is seen that the resistivity should be 17 per cent higher.

A further error of this assumption is that the temperature corresponding to the condenser-neck pressure is not representative of the steam along its path through the interior of the tube bundle, because a drop in pressure and consequently in temperature exists. Consider the tubes inside the condenser exposed to the pressures prevailing at the inlet of the air cooler. The average steam temperature there available is lower due to the pressure drop through the tube bundle; for example, for run No. 65 a temperature drop of 4 F is present, due to the pressure drop from the south wall of the condenser neck to the mean entrance of the air cooler. This difference of four degrees for half the exposed tubes would give 11 per cent difference in resistivity for run No. 65.

Another error is that the surface of the air cooler is included in the total surface of the condenser when the resistivity is evaluated. In the air cooler occurs the lowest temperature rise of the cooling water together with the lowest temperature of the steam, since the air-cooler outlet pressure is always the lowest pressure in the condenser. In most cases these two conditions will have a tendency to offset each other as to their influence on the logarithmic mean temperature difference; but this effect will change with loading, circulating-water inlet temperature, and velocity, and will depend on the size of the air cooler. In this case, although the air cooler amounted to only 8.6 per cent of the total surface of the condenser, the steam-pressure drop through it ranged for all runs from 0.195 to 0.001 in. Hg. The pressure drop through the whole condenser ranged from 0.287 to 0.014 in. Hg.

Some of these effects have been taken into consideration by different corrected formulas based on such items as counter flow, and cross flow,[10] but to evaluate them all is virtually impossible. The author has therefore given some curves showing graphically the influence of this temperature-drop assumption on the apparent resistivity to show the importance of some of these assumptions. This is done in Figs. 11, 12, and 13, in which the deviation of the points at any constant Reynolds number is apparent.

[10] "Some Factors in the Design of Surface Condensing Plant," by H. L. Guy and E. V. Winstanley, Proceedings of the Institution of Mechanical Engineers, vol. 126, 1934, pp. 232–241.

All the possibilities which may require a different temperature drop to be used for the resistivity calculation are not exhausted. It cannot be denied that the inlet temperature is representative of the cooling water entering the condenser. But what about the outlet temperature of the cooling water? In the test of this condenser, an average of 16 thermocouple readings obtained low down in the overflow, was taken as the overflow temperature.

The temperature increase of the cooling water was smaller through the air cooler at the west side of the condenser than through the east side, and temperature differences from west to east of 2 F were observed. The results of these observations are shown in Fig. 14. One must not forget, however, that the temperature gradient directly at the exit of the tubes was greater

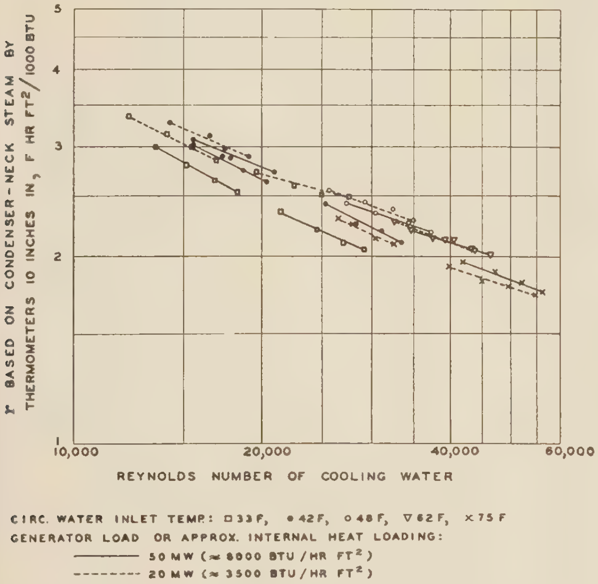

Fig. 11 The Overall Thermal Surface Resistivity r for the Entire Condenser Based on the Steam Temperature at the Condenser Neck

(The straight lines connect four points obtained with different circulator speeds at constant cooling-water inlet temperatures, separate lines being drawn for one-pump and two-pump operation. This resistivity is item 75 of Table 1, and is calculated by dividing a logarithmic mean temperature difference by item 69; it is referred to internal surface. This logarithmic mean temperature difference is based on items 3 and 22, and on the average of items 27 and 28 of Table 1. The abscissas are listed in Table 1 as item 74 and are calculated at the logarithmic mean temperature of the cooling water, item 73.)

because the water was not yet mixed because of the weir.[11] It may here be noted that the more nearly the temperature across the overflow is equalized, the better the performance of the condenser, because all the surface in the condenser will be equally active, and this goes hand in hand with hardly any pressure drop from the condenser neck to the entrance of the air cooler. This is clearly shown in Fig. 14, in which it is seen that the summer runs have a uniform temperature over the full width of the overflow, showing good steam penetration.

To see the effect on the resistivity of the influence of temperature gradient in the overflow water and to eliminate the air-cooler effect, the author has calculated the resistivity for the east half of the condenser with the mean steam temperature of the condenser neck and the hotwell, and has plotted the results in Fig. 15. This was possible because the vertical brace on the center line of the condenser in the overflow, which had no openings, kept the east half of the water circuit separated from the west

[11] "Improved Condenser Design," by C. L. Waddell, *Power Plant Engineering*, vol. 35, 1931, pp. 251–255.

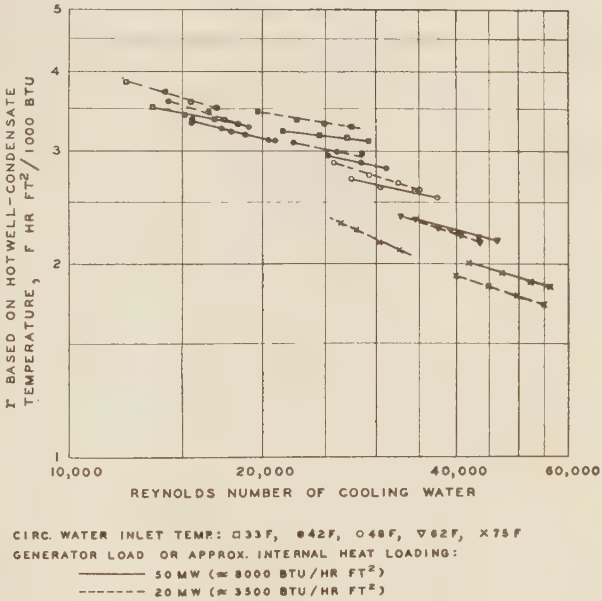

Fig. 12 The Overall Thermal Surface Resistivity r for the Entire Condenser Based on the Hotwell Condensate Temperature

(The straight lines connect four points obtained with different circulator speeds at constant cooling-water inlet temperatures, separate lines being drawn for one-pump and two-pump operation. The resistivity is referred to internal surface. Calculations are similar to those mentioned under Fig. 11, except that for calculating the logarithmic mean temperature difference, item 24 is used instead of the average of items 27 and 28, r in this case being item 76 of Table 1. The abscissas are listed in Table 1 as item 74 as in Fig. 11.)

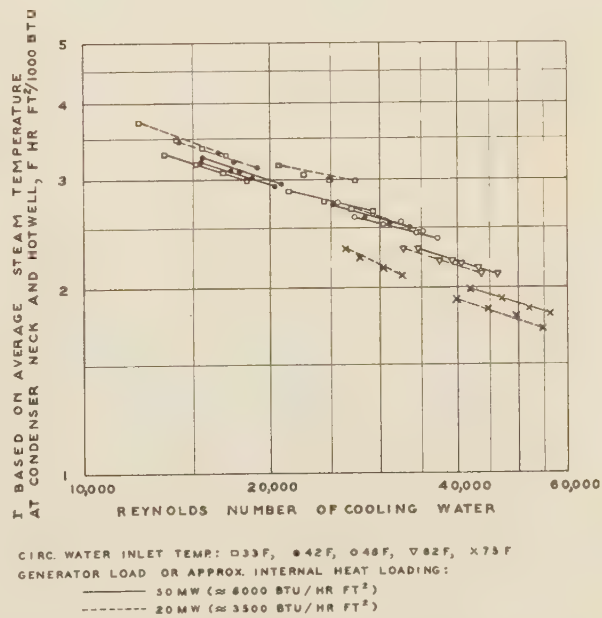

Fig. 13 The Overall Thermal Surface Resistivity r for the Entire Condenser Based on the Average Temperature of the Steam Outside the Tube Banks

(The straight lines connect four points obtained with different circulator speeds at constant cooling-water inlet temperatures, separate lines being drawn for one-pump and two-pump operation. The resistivity is referred to internal surface. Calculations are similar to those mentioned in Fig. 11. Instead of using an average of items 27 and 28 as in Fig. 11, that average has been combined with item 24 to obtain a new average which is used here as the average temperature of the steam outside the tube bank. The abscissas are listed in Table 1 as item 74 as in Fig. 11.)

half. In comparing Figs. 13 and 15, it is seen that this has reduced the scattering of the resistivity considerably, and that, for run No. 65 at 50,000 kw, the resistivity for the east half is 8 per cent less than that of the total condenser.

It is because of all this uncertainty of which temperature to take that the author suggests a simpler way of evaluating the performance of a condenser. The best condenser will have the least resistance to heat flow and the highest condensate temperature at given conditions of inlet temperature and circulator speed. In a modern type of condenser, where there is superheating rather than undercooling of the condensate, the temperature of the condensate will be more representative of the total pressure in the condenser than the pressure at the wall of the condenser neck. Furthermore, both the inlet-water temperature and the condensate temperature can be easily and accurately measured. The author has adopted as a basis for calculating the initial temperature difference, the condensate temperature and the inlet-water temperature.

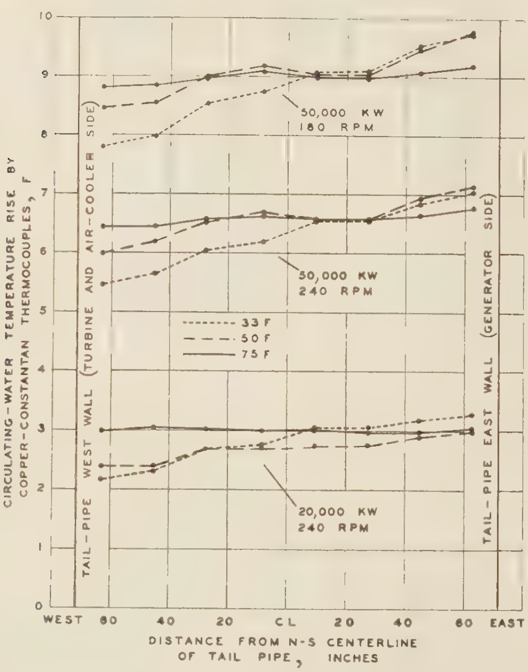

FIG. 14 DISTRIBUTION OF THE CIRCULATING-WATER TEMPERATURE RISE THROUGH THE CONDENSER WITH TWO CIRCULATORS RUNNING AT EQUAL SPEEDS

(Each point represents the average of a north and a south thermocouple located 15 in. apart and equally spaced about the east-west center line at the top of the 30 × 144-in. tail pipe connected to the bottom of the outlet water box of the condenser as shown in Fig. 2. The difference between any north and south thermocouple was always less than 0.2 F because of the thorough mixing by the weir of the water from any vertical row of tubes. The temperatures given are the inlet temperatures of the circulating water. Generator loads and circulator speeds are denominated for each group of curves. These curves are typical of all runs, and show the east-west distribution in the condenser of the heat transferred or amount of steam condensed if it is assumed that the variation of circulating water velocities between different vertical rows of tubes is small from east to west. Note that the west or air-cooler side progressively loses its proper share of the heat load at any speed of the circulators when the circulating-water temperature becomes lower.)

Since the temperature rise of the cooling water is low in a single-pass condenser, the mean heat loading of the whole condenser may be substituted for the heat loading at the cooling-water entrance in calculating the resistivity.

Equation [1c] shows that the resistivity r is a certain temperature difference divided by the heat loading. The fact that this temperature difference is nearly proportional to the heat loading for a constant inlet-water temperature and circulator speed can be very conveniently utilized in checking test results. As an example, Fig. 16 shows this relation taken as a kind of initial temperature difference between the condensate and the inlet-water temperature. Naturally, other temperature differences than that given in this figure can be used; for instance, the steam temperature at the condenser neck minus the inlet or outlet temperature of the cooling water.

The ratio of the initial temperature difference (based on the temperature of the condensate and the inlet water) to the internal-heat loading can be designated as the inlet resistivity. This has been plotted in Fig. 17 as a function of the Reynolds number of the cooling water in the condenser tubes taken at the inlet

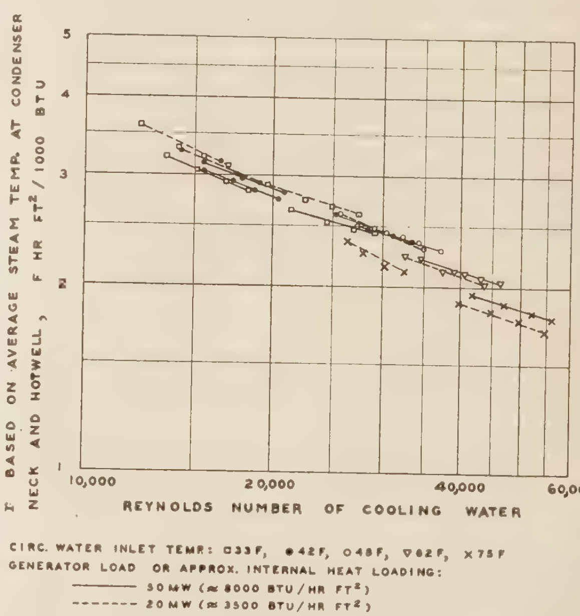

FIG. 15 THE OVERALL THERMAL SURFACE RESISTIVITY r FOR THE EAST HALF OF THE CONDENSER BASED ON THE AVERAGE TEMPERATURE OF THE STEAM OUTSIDE THE TUBE BANK

(The straight lines connect four points obtained with different circulator speeds at constant cooling-water inlet temperature, separate lines being drawn for one-pump and two-pump operation. The resistivity is referred to internal surface. The average steam temperature is that calculated for Fig. 13. The circulating-water temperatures used in calculating the logarithmic mean temperature difference are, however, item 20 instead of item 22, and the average of items 18 and 19, which is item 3 of Table 1.)

temperature. The dashed line of this figure gives a good, simple, and accurate comparison of the all-year performance. The main advantages of this curve are that it is based on measurements which are easily and accurately measured and which are not subject to fluctuations during the course of tests, and that it is directly and conveniently calculated without the use of logarithmic mean temperature differences. The simplicity of this calculation affords the same simplicity in calculations for predicting condenser performance.

To prove the accuracy of predictions made from the dashed curve of Fig. 17, calculations have been made for those runs which deviate most from this curve, using the same conditions of inlet temperature and velocity of the circulating water and heat loading. Table 2 shows that the worst deviation for the temperature of the hotwell condensate is 1.13 F; and for the hotwell steam pressure, the worst deviation is 0.056 in Hg.

CONCLUSION

Results are given of condenser tests with widely varying conditions of internal-heat loading of 3000 to 8000 Btu per hr

TABLE 2 COMPARISON OF CALCULATED AND MEASURED HOTWELL PRESSURE

Assumed conditions:						
Run No. (1)	48	49	61	65	81	88
Inlet-water temperature (3), F	61.30	75.99	76.07	32.97	32.75	33.48
Water velocity through tubes (72), ft per sec	5.04	6.97	4.07	6.91	6.76	2.86
Internal-heat loading (69), Btu per hr and sq ft	3,523	8,165	3,795	7,756	3,411	3,409
Calculations from dash-line of Fig. 17:						
Reynolds number for inlet water	31,700	53,510	31,240	27,440	26,700	11,440
r at inlet from dash-line, Fig. 17	0.00314	0.00216	0.00317	0.00340	0.00345	0.00459
Temperature difference, condensate minus inlet water, calculated, F	11.06	17.63	12.03	26.36	11.77	15.65
Condensate temperature, calculated, F	72.36	93.62	88.10	59.33	44.52	49.13
Hotwell pressure, calculated, in. Hg	0.800	1.590	1.340	0.509	0.295	0.351
Comparison:						
Measured condensate temperature (24), F	71.83	94.65	87.00	60.44	45.46	50.26
Measured hotwell pressure (13), in. Hg	0.783	1.643[a]	1.284	0.548	0.312	0.371
Difference, calculated minus measured condensate temperature, F	+0.53	−1.03	+1.10	−1.11	−0.94	−1.13
Difference, calculated minus measured hotwell pressure, in. Hg	+0.017	−0.053	+0.056	−0.039	−0.017	−0.020

[a] Saturated-steam pressure corresponding to item (23) of Table 1.
NOTE: Numbers in parentheses refer to items in Table 1.

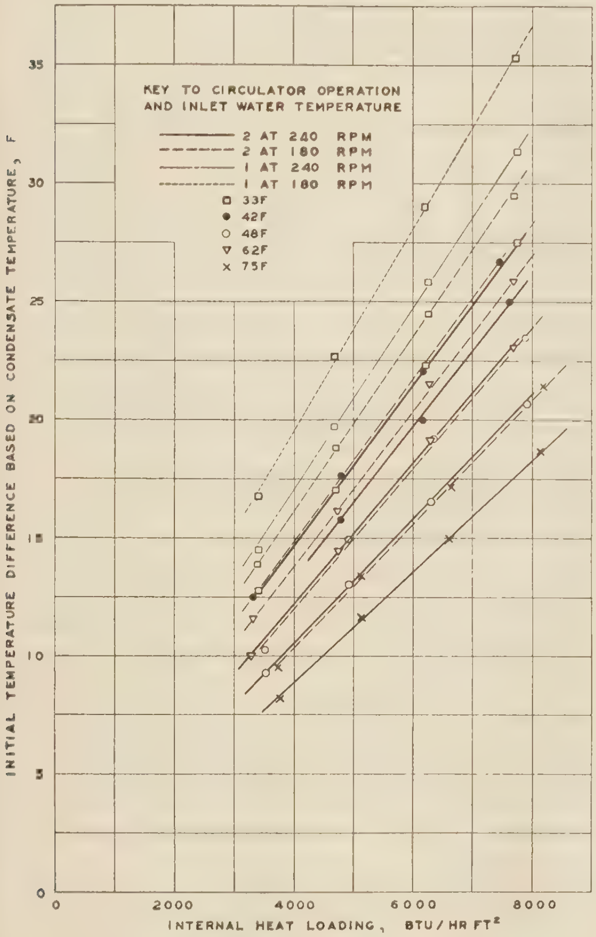

FIG. 16 THE STRAIGHT-LINE PROPORTIONALITY BETWEEN INITIAL TEMPERATURE DIFFERENCE AND HEAT LOADING

(The relation holds for constant conditions of circulating-water inlet temperature and water velocity through the condenser tubes. Instead of constant water velocity, constant nominal conditions of circulator operation have been plotted as designated by the key in the figure. For two-pump operation, the circulator speed of 240 rpm corresponded approximately to a circulating-water velocity of 7 fps in the condenser tubes, while 180 rpm corresponded to 5 fps; and for one-pump operation, 240 rpm corresponded to 4.2 fps, and 180 rpm to 3 fps. The initial temperature difference represented by each ordinate is item 24 minus item 3 of Table 1. The internal-heat loading represented by each abscissa is item 69.)

sq ft, of circulating-water inlet temperature of 32 to 75 F, and of water velocity of 3 to 7 fps. An analysis has clarified two points: (1) The misunderstanding in the superheating of the condensate, and (2) the question of which temperature difference to use in the calculation of the heat-transfer coefficient or resistivity.

1 The superheating of the condensate is due to the partial recovery as pressure of the velocity energy of the steam at the exit of the last wheel of the turbine. It is shown that the condenser-neck wall pressure, on which this superheating of the

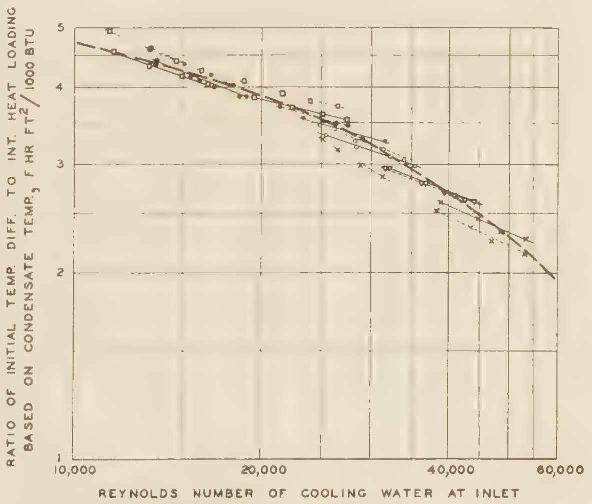

FIG. 17 CORRELATION OF THE RATIO OF INITIAL TEMPERATURE DIFFERENCE TO INTERNAL-HEAT LOADING WITH THE REYNOLDS NUMBER OF THE COOLING WATER AT INLET CONDITIONS

(The heavy broken line indicates the average of all points which can be used for predictions of performance. The straight lines connect four points obtained with different circulator speeds at constant cooling-water temperature, separate lines being drawn for one-pump and two-pump operation. The ordinate is the ratio of the difference between item 24 and item 3 of Table 1 to item 69, and is referred to internal surface. Each abscissa is a Reynolds number based upon item 3 of Table 1.)

condensate is based, is not the correct one to use. Instead, the static plus velocity pressure at the condenser neck, or any other chosen point, should be used.

2 The confusion regarding the heat-transfer coefficient[12] is explained by the fact that the logarithmic mean temperature difference, as generally adopted, is true only for a single spot in the condenser, and is not applicable for the overall heat transfer. This is due to the wide variation of temperature throughout the condenser for the steam as well as for the cooling water.

The heat-transfer coefficient can be replaced by a so-called inlet resistivity which is simpler to calculate and more accurately measured. This inlet resistivity is the ratio of the temperature

[12] "The Surface Condenser," by B. W. Pendred, Isaac Pitman & Sons, London, 1935, pp. 130–136.

difference of the condensate and the inlet water to the internal heat loading (which is defined as the heat transferred at the inside surface of the tubes per unit area and time). This inlet resistivity plotted as a function of the Reynolds number based on the water-inlet temperature gives a good curve to establish the all-year-around performance of a condenser.

All statements are substantiated by test figures.

ACKNOWLEDGMENTS

The opportunity afforded the author by The Detroit Edison Company for such extended research work should receive special acknowledgment. The author also wishes to recognize the efficient assistance rendered him by his associate, J. D. Starkweather, in conducting the experiments and calculating the results.

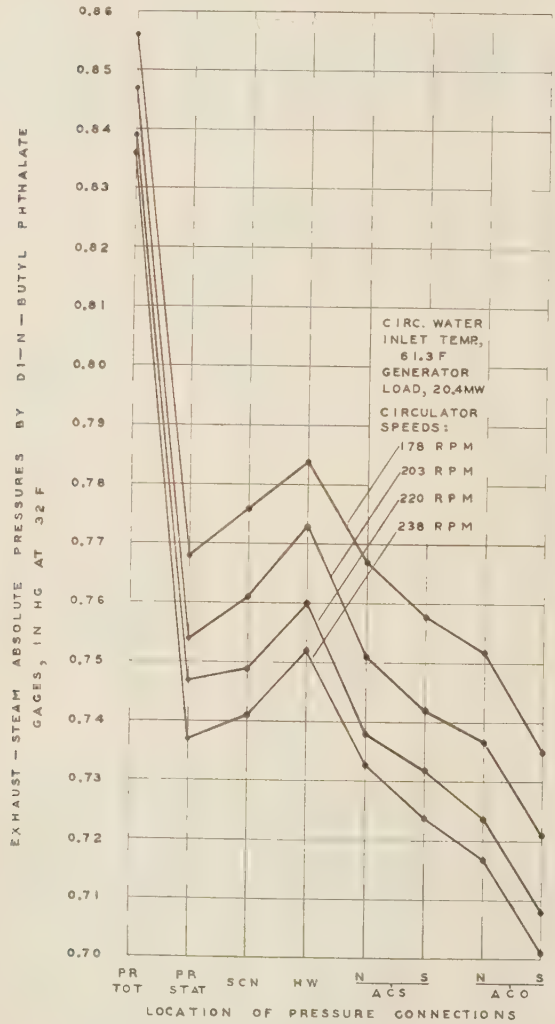

Fig. 18 Typical Example of the Systematic Behavior of the Absolute-Pressure Oil-Gage Indications

(The different points at which pressure measurements were made are, reading from left to right on the abscissa, the corrected total pressure at the Prandtl tube, item 53; the corrected static pressure at the Prandtl tube, item 54; the pressure at the south wall of the condenser neck, item 12; the pressure at the west wall of the hotwell, item 13; and so on in succession, items 14 to 17 of Table 1, the condenser end of the dry-vacuum-pump suction line being the same as the south end of the air-cooler outlet, that is, *ACO, S*. It should be noted that the magnitude of the differences between each of these curves is only a few thousandths of an inch of mercury.)

Appendix

Details of Principal Measurements

Exhaust-steam pressures were measured at eight points, as indicated in Table 1 and Fig. 2, by absolute-pressure oil gages, which were filled with di-n-butyl phthalate. These gages are sensitive and accurate to 0.003 inch Hg. In Fig. 18, which shows a typical example of the systematic behavior of the gages, it should be noted that the magnitude of the differences between each of the curves is only a few thousandths of an inch of mercury. The oil gages have been described in another article.[13] Fig. 2 of that article is an illustration of the arrangement for this condenser test.

The leakage of the gage lines and connections was negligible; for example, the hotwell-gage line, which had a volume of 69 cu in., showed pressure-rise rates of the order of 0.01 in. Hg per hr in a leakage test with an absolute pressure of 0.08 in. Hg in the line.

To measure the exit-steam velocity of the turbine, a Prandtl-type pitot tube[14] was used. The nose of this Prandtl tube was only 6 in. downstream from the exit face of the lower part of the twenty-first wheel of the turbine. The axis of the tube was parallel to the axis of the turbine and it was located in the south half of the turbine exhaust hood in a plane through the turbine axis making an angle of about 33 deg with the vertical, and at a radius of 70 in. from the turbine axis.

Thirty-two copper-constantan thermocouples, accurate within 0.1 F, were directly exposed to measure various temperatures. Their cold junctions were kept in an ice bath, and they were used with a Leeds and Northrup type-*K* potentiometer, with which changes of 0.05 F can be estimated from graduations equivalent to 0.25 F. Before the condenser tests, the thermocouples were calibrated in a stirred-water bath at seven or thirteen points between 39 and 100 F; and after the condenser tests, or one year later, they were calibrated at four points between 40 and 80 F and showed no change. Parabolic calibration curves evaluated by the method of least squares fitted the calibration data very accurately throughout this range.

The accuracy of the thermocouple setup was kept within 0.1 F through attention to sources of error such as effects of nonhomogeneity of the thermocouple lead wires, zero shift of the galvanometer, and thermal electromotive forces of the selector switch. A check on the accuracy of the thermocouple setup in the power plant was possible by a comparison of the circulating-water inlet temperature as measured by one of the thermocouples in the west circulator discharge line and a calibrated mercury thermometer in a well in this line. Results of some typical runs are given in Table 3.

TABLE 3 COMPARISON OF A MERCURY THERMOMETER IN WELL AND AN EXPOSED COPPER-CONSTANTAN THERMOCOUPLE

Run No.	Circulating-water-inlet temperature, F — Thermocouple No. 160	Mercury thermometer
81	32.7 F	32.9
68	33.3	33.4
6	42.5	42.7
26	50.2	50.3
40	63.3	63.2
52	75.5	75.7

Six of the thermocouples were put in the condenser neck, spaced at the center of the areas between the braces, and one was mounted at the Prandtl tube. As originally installed, these were

[13] "An Oil Manometer," by G. H. Van Hengel and J. D. Starkweather, *Mechanical Engineering*, vol. 57, October, 1935, pp. 633–635.

[14] "Messung Strömender Luft mittels Staugeräten," by H. Kumbruch, *Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 240. 1921, Figs. 18 and 19, p. 10.

TABLE 4 COMPARISON OF STEAM TEMPERATURES AT CONDENSER NECK WITH 33 F INLET WATER

Run No.	65	72	81	88
Generator load, kw	49,680	50,030	20,540	20,640
South wall:				
By oil gage, F	54.3	65.4	43.7	49.0
By thermometer, F	54.5	65.6	44.1	49.7
South thermometers 9 in. in:				
At center line, F	53.2	64.4	43.0	48.7
19 inches East of center line, F	53.2	64.6	43.2	48.6
Shielded thermocouples:				
East half:				
South section, F	55.5	64.8	43.2	48.5
Middle section, F	55.5	65.5	43.5	49.1
North section, F	51.4	63.5	43.2	47.8
West half:				
South section, F	51.9	63.7	42.1	47.7
Middle section, F	52.6	63.8	42.5	47.7
North section, F	51.9	63.4	42.1	47.6
North thermometer 9 in. in, F	51.3	63.9	42.5	48.3
North wall by thermometer, F	54.6	65.8	44.1	49.4

TABLE 5 COMPARISON OF EXHAUST-STEAM TEMPERATURES AT PRANDTL TUBE BEHIND LAST WHEEL OF TURBINE

Run No.	81	84	85	88
From corrected static pressure of Prandtl tube, F.	44.3	44.8	45.9	48.9
Shielded thermocouple, F	42.9	43.5	45.3	48.6
Grounded thermocouple, F	46.1	46.4	46.9	49.7

directly exposed. Erratic readings resulted because of considerable charges of static electricity deposited on the junctions of these thermocouples, probably by the moisture in the steam. Sufficient current flowed through their lead wires to cause errors in the readings of as much as 4 F. Subsequent grounding of the thermocouple junctions failed to give correct readings and finally for the last series of tests, at inlet-water temperature of 33 F, they were covered with grounded shields with the exception of the thermocouple at the Prandtl tube. An additional shielded thermocouple was installed at this point to compare it with the grounded arrangement. Tables 4 and 5 illustrate the results obtained in various runs of the last series.

The comparison of the hotwell steam pressure and temperature measurements furnish a good example of a check which greatly strengthens the data obtained by the oil-gage and thermocouple setups. Table 6, which contains data from a few runs, is typical of the results obtained.

TABLE 6 COMPARISON OF HOTWELL STEAM OIL-GAGE, AND THERMOCOUPLE READINGS

Run No.	81	88	6	26	40	52
Oil gage no. 3 (hotwell steam):						
Readings, cm $C_{16}H_{22}O_4$[a]	10.34	19.85	24.34	29.28	43.16	58.32
Room temperature, F	79.5	82.2	91.8	81.7	89.5	92.3
Thermocouple no. *T*165 (hotwell steam):						
"Normal" readings, mv	0.2985	0.684	0.813	0.941	1.206	1.430
"Reverse" readings, mv	0.3015	0.689	0.817	0.947	1.212	1.433
Average, mv	0.300	0.687	0.815	0.944	1.209	1.432
Hotwell steam pressure:						
By oil gage no. 3, in Hg at 32 F	0.312	0.598	0.730	0.882	1.295	1.748
By thermocouple no. *T*165, in. Hg at 32 F	0.312	0.595	0.728	0.885	1.298	1.756
Hotwell steam temperature:						
By oil gage no. 3, F	46.0	63.9	69.6	75.2	87.0	96.7
By thermocouple no. *T*165, F	46.0	63.8	69.6	75.3	87.1	96.9

[a] Di-n-butyl phthalate.

TABLE 7 VACUUM CORRECTIONS FOR EXPOSED MERCURY THERMOMETERS

Absolute pressure, in. Hg	0.2	0.5	0.9	5.1
Vacuum corrections, F:				
Thermometer no. 42779	+0.33	+0.28	+0.27	+0.18
Thermometer no. 40347	+0.18	+0.13	+0.12	+0.08

Steam temperatures at five locations in the condenser neck were measured with directly exposed mercury-in-glass thermometers. The results of these measurements are given in Table 4. (See Fig. 2.) Direct exposure was accomplished by inserting the thermometers through rubber stoppers, all joints being sealed by the liberal application of jellied castor oil.

In addition to the conventional calibration in a stirred-water bath, a special calibration was made on the thermometer bulbs for the effect of vacuum. In the latter calibration, the thermometers and a calibrated copper-constantan thermocouple were installed to full immersion with direct exposure in a vacuum-tight and thermally insulated container, which had been filled with transformer oil at about 75 F. The corrections due to vacuum only are given in Table 7 for the extreme cases of fifteen thermometers:

Physical-Property Uniformity in Valve-Body Steel Castings

By A. E. WHITE,[1] C. L. CLARK,[2] ANN ARBOR, MICH., and SABIN CROCKER,[3] DETROIT, MICH.

This paper describes an extensive investigation of the physical properties of specimens cut from various sections of large welding-end valve bodies made of carbon-molybdenum cast steel. Macrosections, impact values, and density determinations are presented in addition to the usual tensile properties.

The results indicate that the welding ends are the soundest part of the valve body and that sections cut from the valve body as close as possible to the welding ends have physical properties commensurate with those obtained from conventional separately and integrally cast test bars.

Evidence of rather general porosity was disclosed by the macrosections from the thicker-walled portions of the bodies and in the bonnet flange, with consequent low ductility values for some of the tensile specimens from these locations. While this condition was considered fairly acceptable when allowance was made for the greater metal thickness in these sections, the investigation showed these to be the regions where improvements could be effected through further development of casting procedure.

HIGH-GRADE steel castings for valve bodies are purchased customarily on the basis of specifications, in which are included requirements for chemical composition and for tension, bend, and hydrostatic tests. Sometimes impact and X-ray tests are required, and in exceptional cases, metallographic standards are incorporated. Tensile specimens to check conformance with specified values usually are machined from lugs cast attached to the valve bodies when of sufficient size, otherwise from separately cast test bars. In only a few cases, so far as the authors know, does a specification call for tension-test specimens from the body of a casting. One of these is in a specification written by a bureau of the U. S. Navy Department where a large number of castings of a similar type were purchased at one time. In such instances the destruction of a casting for test purposes is entirely feasible, although ordinarily the cost would be excessive. In any event such destructive testing can only serve to supplement information obtained from conventional test bars representing each casting.

It has been assumed that values obtained from tension tests made on specimens cut from separately or integrally cast lugs are reasonably representative of the properties which would be found in the bodies of the castings themselves. This assumption, of course, presupposes that due allowance is made for the influence of mass effect and design. While little information has been published on the relation of tensile-test results obtained from test bars as distinguished from the casting proper, reference is made to three recent surveys which touch on this subject.[4,5,6]

With such a background The Detroit Edison Company felt it desirable, in rebuilding its Conners Creek power plant for higher pressures and temperatures, to obtain assurance that the soundness of welding-end valve bodies was such that they could be used safely in the superheated-steam and boiler-feed piping. The major change in the design of the valve bodies from that previously used is the elimination of the end flanges and the use of welding ends in which the wall thickness approaches that of the adjoining pipe. The extent to which foundry technique had successfully followed this change in design by providing adequate and properly placed gates and risers, therefore, called for a comprehensive examination of actual valve-body castings.

Accordingly the threefold purpose of this investigation, which was made under the sponsorship of The Detroit Edison Co. and with the assistance and cooperation of a number of valve-manufacturing concerns, was to learn through sectioning full-sized valve bodies: (1) Whether sound cast material could be expected in welding ends with their reduced metal sections; (2) the extent to which the physical properties obtained on sections cut from the body, as close as possible to these ends, would differ from results obtained on lugs either separately or integrally cast; and (3) the uniformity of the physical properties throughout an entire valve body.

Discussion of Results

While the results showed a considerable variation, they were most encouraging and gratifying, especially with respect to those sections on which it is believed emphasis should be placed.

The macrographs showed that the welding ends of all of the castings appeared to be sound. However, the physical tests disclosed certain differences. One of the castings met the re-

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Contributed by the Power Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

[4] "Specifications for Chrome-Tungsten Steel Castings," by B. B. Westcott, V. T. Malcolm, and H. Henderson, *Refiner and National Gasoline Manufacturer*, vol. 12, July, 1933, p. 281.

[5] "Cast Metals Handbook," American Foundrymen's Association, 222 West Adams Street, Chicago, Ill., 1935 edition, p. 333.

[6] Symposium on Steel Castings, Proceedings of the American Society for Testing Materials, vol. 32, 1932, part 2, technical papers, p. 43. Particular attention is called to "Purchase Requirements for Steel Castings With Notes on Physical Properties in Cast Bars and in Commercial Castings," by R. A. Bull, p. 77.

TABLE 1 CHEMICAL COMPOSITION OF FOUR C-Mo CASTINGS

Casting	Chemical composition, per cent: C	Mn	Si	S	P	Mo
A	0.30	0.75	0.33	0.011	0.020	0.51
B	0.28	0.89	0.45	0.035	0.022	0.57
C	0.25	0.83	0.39	0.022	0.033	0.47
D	0.26	0.73	0.36	0.020	0.030	0.48

NOTE: Specified chemical requirements are C = 0.20–0.35, Mn = 0.70–1.00, Si = 0.30–0.50 S = 0.05 max, P = 0.05 max, and Mo = 0.40–0.60.

TABLE 2 DENSITY TESTS ON SPECIMENS FROM THREE C–Mo CASTINGS

Designation	Description	Casting *A*	Casting *B*	Casting *C*
S	Separately cast	7.86	7.80	7.824
I	Integrally cast	7.86	7.81	7.822
*C*1	Bonnet flange, cope	7.86	7.80	7.827
*C*2	Bonnet flange, drag	7.86	7.81	7.818
C3-L	Welding end, cope	7.83	7.82	7.816
C4-L	Welding end, drag	7.85	7.80	7.822
C5-L	Shell, cope	7.86	7.80	7.828
C6-L	Shell, drag	7.87	7.81	7.816

quirements for soundness perfectly, two others were passable and it is believed that with changes in gates and risers the fourth could be made to be satisfactory. In view of these findings, therefore, there is no question but that sound cast material can be obtained in the welding ends of valve-body castings.

The physical tests showed that it is possible to get nearly as acceptable properties in the welding-end sections as it is from sections taken from separately or integrally cast bars. This statement is borne out by the results obtained from three of the castings, one of which was most satisfactory and two others passably so. These results were most reassuring not only in that the welding ends of all of the bodies examined were found to be the soundest part of the castings, but also in demonstrating that the same physical properties might reasonably be expected from

TABLE 3 TENSILE PROPERTIES OF SPECIMENS FROM FOUR C-Mo CASTINGS

Specimen designation	Description	Property	Casting *A* Spec. 1	Casting *A* Spec. 2	Casting *B* Spec. 1	Casting *B* Spec. 2	Casting *C* Spec. 1	Casting *C* Spec. 2	Casting *D* Spec. 1	Casting *D* Spec. 2
		Separately cast coupon tests								
CM-S	Separately cast	TS	82100	...	88900	82950	80875	79000	71000	72450
		YP	57300	...	61200	54200	52500	51750	47500	55650
		El in 2 in.	27.5	...	23.6	22.0	31.3	32.0	27.5	25.0
		R of A	51.9	...	51.4	42.5	62.7	60.0	48.9	44.0
Average of tests from separately cast coupons		TS	82100		85925		79938		71725	
		YP	57300		57700		52125		51575	
		El in 2 in.	27.5		22.8		31.7		26.3	
		R of A	51.9		47.0		61.4		46.5	
		Integrally cast coupon tests								
CM-I	Integrally cast	TS	82200	...	85850	83600	...	...	71900	...
		YP	58800	...	58500	53300	...	..	47450	...
		El in 2 in.	28.0	...	26.5	27.5	...	...	27.5	...
		R of A	53.3	...	48.3	51.9	...	...	48.9	...
Average of tests from integrally cast coupons		TS	82200		84725		...		71900	
		YP	58800		55900		...		47450	
		El in 2 in.	28.0		27.0		...		27.5	
		R of A	53.3		50.1		...		48.9	
		Welding-end section tests								
C3-L	Long., cope, welding-end	TS	80600	78650	81000	76200	75300	77500	71300	...
		YP	55300	51500	48900	47450	48700	58000	44700	...
		El in 2 in.	27.0	25.5	20.7	24.0	22.0	26.0	16.3	...
		R of A	36.0	45.8	34.1	43.4	22.2	45.5	27.9	...
C4-L	Long., drag, welding-end	TS	80600	77750	83000	81150	75000	78000	70150	...
		YP	56900	52100	52500	50350	47750	59500	45550	...
		El in 2 in.	24.0	22.5	21.8	23.0	20.5	24.5	15.8	...
		R of A	47.5	50.6	39.4	45.5	33.5	33.1	23.6	...
Average of tests from welding-end		TS	79400		80338		76450		70725	
		YP	53950		49800		53488		45125	
		El in 2 in.	24.8		22.4		23.3		16.1	
		R of A	45.0		40.6		33.6		25.8	
		Shell-section tests								
C5-L	Long., cope, shell	TS	77300	76350	72500	70600	74875	80000	...	...
		YP	50100	47350	45400	42750	47500	49500	...	...
		El in 2 in.	23.0	22.5	22.6	23.5	26.5	19.0	...	...
		R of A	36.3	27.8	36.6	37.3	43.4	27.2	...	...
C6-L	Long., drag, shell	TS	78600	79300	78300	75700	...	...	...	...
		YP	50240	49050	47500	45200	...	...	...	...
		El in 2 in.	20.0	20.5	13.3	14.5	...	...	...	...
		R of A	26.8	35.0	20.0	21.6	...	...	...	...
Average of tests from shell		TS	77888		74275		77438		...	
		YP	49185		45213		48500		...	
		El in 2 in.	21.5		18.5		22.8		...	
		R of A	31.5		28.9		35.3		...	
		Bonnet-section tests								
C1-T	Tang., cope, bonnet	TS	78200	77950	83800	81900	78625	80000	...	...
		YP	52000	51000	51750	54050	51300	50000	...	...
		El in 2 in.	16.0	21.0	16.0	12.5	18.5	25.5	...	...
		R of A	20.6	29.2	26.8	16.3	26.8	36.6	...	...
C2-T	Tang., drag, bonnet	TS	80800	78700	84700	82200	77260	79000	...	...
		YP	55900	51400	53200	50750	48300	49000	...	...
		El in 2 in.	28.0	29.0	25.7	25.0	20.0	30.5	...	...
		R of A	49.2	38.8	44.9	41.9	49.4	58.6	...	...
C1-R	Radial, cope, bonnet	TS	84800	77850	84150	82900	77500	79000	...	...
		YP	52300	50750	51500	53350	49600	49500	...	...
		El in 2 in.	19.0	15.5	16.3	12.0	18.5	21.0	...	...
		R of A	23.7	24.4	24.8	17.0	30.9	25.4	...	...
C2-R	Radial, drag, bonnet	TS	80800	78650	82200	81400	76750	81250	...	...
		YP	56900	50850	47900	48200	48500	50500	...	...
		El in 2 in.	25.0	28.0	19.0	21.0	24.5	30.5	...	...
		R of A	38.8	43.4	30.0	18.4	35.4	50.0	...	...
Average of tests from bonnet		TS	79719		82906		78673		...	
		YP	52638		51340		49613		...	
		El in 2 in.	22.7		18.7		23.6		...	
		R of A	33.5		27.5		39.0		...	
Average for entire set of values		TS	79511		81150		78121		71360	
		YP	52763		50898		50744		48170	
		El in 2 in.	23.4		20.5		24.4		22.4	
		R of A	37.7		34.7		40.0		38.7	

NOTE: Specified minimum values for separately or integrally cast coupon specimens are: Tensile strength = 70,000 lb per sq in., yield point = 50,000 lb per sq in., elongation in 2 in. = 20 per cent, and reduction of area = 35 per cent.

Abbreviations: Tang. = tangential; long. = longitudinal; spec. = specimen; TS = tensile strength, lb per sq in.; YP = yield point, lb per sq in.; El in 2 in. = elongation in 2 in., per cent; and R of A = reduction of area, per cent.

sections cut near the welding ends as from conventional test bars.

The results were sufficiently encouraging to lead to the further conclusion that where reasonable physical requirements are specified, test sections from any portion of this type of casting can be secured which will comply with the minimum requirements. In order to meet these requirements, it might in some cases be necessary for the steel-casting companies to experiment for the purpose of obtaining the right location for gates and risers, the proper pouring temperature, and the most effective heat-treatment procedure. In comparing results, due consideration must be given to the necessity for a heat-treatment procedure which will place the specimens from sections of unequal thickness in as nearly the same structural condition as possible.

Fig. 1 Macrograph From Welding End and Shell of the Cope Side of a Casting

Test Data and Their Analysis

In carrying out the investigation, a number of valve companies provided four carbon-molybdenum cast-steel valve bodies and test lugs to go with them from which the coupons for tensile, impact, and density tests were secured, as well as the sections for macrographic examination. Care was taken to secure similarly located sections from each of the castings. These castings were made for 8-in. and 10-in. valves of the 900-lb standard and weighed 800 to 1200 lb each. About half of the tension and impact tests were made by the valve companies. The remainder of the tension and impact tests, the density tests and the macrographs included in this paper were made at the University of Michigan.

Chemical Composition. The chemical composition of these carbon-molybdenum castings, as determined by the various companies furnishing the castings, is given in Table 1. It shows all of the castings met the given requirements.

Density Tests. Density tests were made on samples from sections of three of the four castings. It was not expected that a great deal of difference would be found in the various density determinations, yet it appeared that such determinations would be an indication of the soundness of the metal. The test fails in this purpose because small sections must be used for measurement. These small test coupons seldom contain the blowholes, shrink cavities, or sand spots present elsewhere in the casting.

The test is interesting, however, in that it shows casting *A* is somewhat more dense than the other two castings from which density tests were made. This checks with the findings of the tensile and impact tests.

The data setting forth these findings are given in Table 2.

Fig. 2 Macrosection From the Drag Side of Bonnet Flange

Tensile Tests. The results of the tensile tests are given in Table 3. These tests are relatively complete for three of the castings, but only a few tests were made on the fourth.

It is not easy to evaluate these results properly, because many of the test specimens were taken from sections wherein varying degrees of soundness, with resultant changes in physical properties, are to be expected unless foundry technique is relatively perfect.

On the basis of general averages, three of the four castings met the given tensile requirements. The fourth would have met them by a slight modification of the heat-treatment. It is not always safe, however, to base the acceptability of a casting on general averages for with but one serious flaw or one unduly low physical property, the casting would be considered unfit for the purpose for which it was intended.

Most physical tests made to determine the acceptability of castings are taken from coupons cut either from separately cast or integrally cast bars. There is no reason why any casting should be accepted, where reasonable specifications are used, if test specimens taken from conventional bars do not meet the requirements. In this case the results of tensile tests on conventional bars from all the castings were essentially as specified.

In view of the fact that the valve bodies were designed with welding ends where the metal thickness of the casting was reduced to approximately that of the pipe to which these ends were to be welded, it was highly important that the material should meet fully the physical requirements given in the specifications. The tests showed casting *A* did, castings *B* and *C* virtually did, but such was not the case for casting *D*. Casting *D* had too low an elongation and reduction of area, showing that the casting was insufficiently sound in this section.

The shell-section tests showed castings A, B, and C to be quite satisfactory generally, particularly when account is taken of metal thickness. No tests were made on shell-sections from casting D. Casting A was only slightly low in yield point and reduction of area. Casting B was lower than need be in elongation and reduction of area and slightly low in yield point. Casting C, which was tested only on the cope side, was slightly low in yield point and, in one of the tests, in elongation and reduction of area. What would have been the result if tests had been made on the drag side of this casting must be left to speculation.

Fig. 3 Macrosection From the Drag Side of Shell

The tests on the bonnet-flange sections showed a fairly acceptable condition for the three castings examined, namely, castings A, B, and C. While the ductility was not all that it might have been in the cope side of both the tangential and radial sections, this is not cause for undue alarm. Beyond question, low ductility properties have existed in these sections since castings of this type have been made. It is believed, however, that with further effort, the ductility values could be raised to meet the minimum values called for in the specifications from integrally or separately cast test bars.

The yield point also was slightly low in certain of the sections from castings B and C, but not sufficiently so to warrant adverse comment. From the tests on these sections casting C appears slightly superior to casting A, and casting A appears slightly superior to casting B.

Impact Tests. Table 4 gives the results of the Charpy impact test using the type of V-notch specimen described in A.S.T.M. Proceedings, Vol. 34, 1934, Part I, p. 1204, Fig. 1(a). On the basis of averages, all of the castings met the given requirements on the sections taken from separately or integrally cast bars. The tests on the welding-end sections showed the metal in casting A to be satisfactory, that in castings C and D the metal was practically satisfactory, and that in casting B the metal evidenced need for improvement.

The shell-section impact tests show a generally satisfactory condition. It is interesting to note that whereas castings A and B get their best values from the drag side, casting C gets its best values from the cope side. The tests from the bonnet section show casting A uniformly acceptable, casting B passably so, and casting C in need of improvement.

Attention is called to the fact that all of the impact values from casting A, except those from the shell sections, were markedly superior to those from the other castings. Also, the average of the impact values of casting A was well above the minimum called for in the specifications from coupons when taken from separately cast or integrally cast lugs.

Macro Examination. As a means of examining the various sections for soundness, numerous macro examinations were made on deep-etched specimens cut from the various valve bodies. Since the acid etch accentuates segregations and physical discontinuities, it reveals defects not evident in cut or fractured sections. The four macrosections shown in Figs. 1 to 4, inclusive, are representative of the various types of cast structure found in the valve bodies.

Fig. 4 Macrosection From the Cope Side of Welding End

Fig. 1 shows a macrosection taken from the welding end and shell of one of the castings examined. Because of its small cross-sectional area when compared with that of the section taken from the drag side of the bonnet, it shows a relative absence of dendritic structure. The blowholes are also less pronounced. The section, however, shows a crack at rather a critical part of the casting and, therefore, a casting with such a crack in such a location is not acceptable.

Fig. 2 shows a typical macrosection of one of the castings taken from the drag side of the bonnet flange. Dendrites are present, although not markedly so. Porosity due to shrinkage is also present. The section is free from cracks. This metal structure may be viewed as more or less typical of that usually found in such sections. It is not an ideal condition; the metal is too porous and many of the blowholes are too large.

Fig. 3 shows a typical structure of one of the castings taken

TABLE 4 CHARPY IMPACT PROPERTIES OF SPECIMENS FROM FOUR C-Mo CASTINGS, FT-LB

Specimen designation	Description	Casting *A*	Casting *B*	Casting *C*	Casting *D*
	Separately cast coupon tests				
CM-S	Separately cast	41.0	30.5	18.0	32.0
		40.0	35.0	33.0	28.0
		..	28.5	24.0	25.0
		..	44.0	24.0	25.0
		..	28.0	41.0	32 0
		..	22.0	..	..
		..	26.0	..	..
	Average	40.5	30.6	28.0	28 4
	Integrally cast coupon tests				
CM-I	Integrally cast	59.0	37.0	..	26
		63.0	43.0	..	26
		..	42.0	..	23
		..	..	..	28
	Average	61.0	40.7	.	25.8
	Welding-end section tests				
C3-L	Long., cope	41.0	19.0	23.0	29.0
		27.0	18.0	20.0	23.5
		43.0	16.0	24.0	..
		32.0	13.0	24.0	..
C4-L	Long., drag	54.0	23.5	20.5	22.5
		40.0	20.0	20.5	22.0
		53.0	23.0	23 0	..
		41.0	21.0	21.0	..
		..	..	20.5	..
C3-T	Tang., cope	48.0	16.5	24.0	25.0
		33.0	20.0	22.0	28.0
		31.0	17.0	20.5	24 0
		..	..	20.5	.
C4-T	Tang., drag	60.0	22.0	24.0	20.0
		47.0	21.0	25.0	23 0
		46.0	23.0	23.0	26 0
		..	..	24.0	.
	Average	42.6	19.5	22.3	24 3
	Shell-section tests				
C5-L	Long., cope	20.0	29.5	22.0	..
		28.0	22.5	22.0	.
		20.0	21.0	27.5	.
		18.0	16.0	33.5	.
C6-L	Long., drag	28.0	21.5	17.0	.
		32.0	27.0	19.0	.
		29.0	21.0	26.0	.
		29.0	25.0	24.0	.
	Average	25.5	22.4	23.9	..
	Bonnet-section tests				
C1-R	Radial, cope	28.0	25.0	24.0	..
		28.0	27.5	16.0	..
		24.0	20.0	24.0	..
		28.0	24.0	26.5	..
C2-R	Radial, drag	33.0	26 5	20.0	..
		36.0	19 5	16.0	..
		35.0	23 0	20.5	..
		34.0	18 0	20.5	..
	Average	30.8	22.9	20.9	..
Average for entire set		36.7	24.4	23.1	25.7

NOTE: Specified from separately or integrally cast sections = 25 ft-lb minimum. Tests made on A.S.T.M. V-notch specimen.

from the shell on the drag side. It shows evidence of a dendritic structure, although it is not marked. Blowholes also are present. Likewise, it discloses a considerable degree of porosity. There are no cracks, however, and in general the structure disclosed is acceptable.

Fig. 4 shows a macrosection of a casting taken at the welding end of the cope side. It represents excellent structure. There are few, if any, dendrites and the metal is quite sound and free from cracks. The apparent line at the top of the section is not a crack, but is the result of small blowholes or metallic inclusions. The general structure represents an ideal. It has been possible to obtain it because of good casting practice and also because of the nature, shape, and cross-sectional area of the part in question. In other locations, or with metal of greater wall thickness, it might prove to be more difficult to get a section as sound as the one shown in Fig. 4.

Summary

This paper gives the results of tests made on carbon-molybdenum cast-steel valve bodies with welding ends to determine (1) if the material in the welding ends is sound; (2) if the properties in these welding ends differ from those in separately or integrally cast lugs; and (3) if the physical properties throughout the entire valve bodies are reasonably uniform.

The results showed that for valve-body castings of the type examined it is possible to obtain: (1) Sound material in the welding ends; (2) properties from welding ends commensurate with those from conventional bars; and (3) castings with reasonably uniform physical properties. These conclusions are stated with due recognition of the fact that the tensile and impact results were not in every case entirely acceptable. However, the results were of such a nature as to lead to the conclusion that after further study with respect to the proper location and size of gates and risers, proper pouring temperature, and determination of the most suitable heat-treatment, sections could be secured from any location which would uniformly meet the given tensile and impact requirements set forth in this paper. In making this statement it is recognized that the goal proposed is not necessarily one which can be attained at once, since a considerable amount of experimentation in foundry practice might be required.

Experimental Determinations of the Flow Characteristics in the Volutes of Centrifugal Pumps

By R. C. BINDER, LAFAYETTE,[1] IND., and R. T. KNAPP,[2] PASADENA, CALIF.

This paper is a discussion of an exhaustive study which was made to determine internal-flow characteristics of two high-efficiency, high-head centrifugal pumps of commercial design. Special equipment was constructed to measure instantaneous values of pressures and velocities at various stations in the volutes. The authors discuss previous work of a nature similar to their study. They describe briefly the apparatus used in their tests, the arrangement of the test equipment, and the accuracy of the equipment used to measure the instantaneous values of pressures and velocities in the volutes. The results of the tests are presented in graphical form.

Background and Objectives

THE DESIRE for a knowledge of actual flow conditions in the impellers and volutes of centrifugal pumps has long been present in the minds of those interested in the construction or operation of hydraulic machinery. This desire has grown more intense as efficiencies have been forced higher and higher, since each increase in efficiency has been more difficult to attain, and has demanded more precise information about the hydraulic behavior within the casing of the machine. Therefore, it is not surprising that many attacks have been made on this problem, both from the theoretical and the experimental points of view. However, as yet neither method of approach has yielded entirely satisfactory results, so that additional attempts to supply this knowledge are still definitely needed.

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[2] Assistant Professor of Mechanical Engineering, California Institute of Technology. Jun. A.S.M.E. Professor Knapp was graduated with a B.S. degree from the Massachusetts Institute of Technology in 1920, and received his Ph.D. degree in mechanical engineering from the California Institute of Technology in 1929. He was designer for the Gay Engineering Corporation 1920-1921, and instructor at the California Institute of Technology from 1922 to 1930. Since 1923 he has been in charge of the power-plant laboratory at the California Institute of Technology, and has been in charge of the hydraulic laboratory since 1927. He acted as consulting engineer for the Riverside Cement Company, Los Angeles, Calif., during 1929 and 1930. At present he is also consulting engineer for the Metropolitan Water District of Southern California.

Contributed by the Hydraulic Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Theoretical Treatments. Many of the theoretical studies have started with the assumptions of perfect fluids and potential flow. The works of Kucharski, (1)[3] von Busemann, (2) Schultz, (3) Sörenson (4), and Uchimaru and Kito (5) are in this class. Kucharski treated mathematically the problem of an impeller with straight radial vanes. Spannhake (6) pointed out that fluid passages, formed by curved vanes of finite length and cut off by entrance and exit circles, as found in actual practice, present many difficulties to theoretical investigators. It is well known that the actual values of both the magnitude and direction of the absolute exit velocity do not agree with those calculated on the basis of potential flow. Pfleiderer (7) therefore calculated the theoretical head developed by a pump on the condition that the relative exit angle was less than the vane angle. Fischer and Thoma (8) concluded that: "Practically all flow conditions for an actual fluid are fundamentally different from those theoretically derived for an ideal frictionless fluid."

It should be noted that the treatments just mentioned refer almost entirely to the flow within the impeller and give few or no data on the action within the volute. This is decidedly a shortcoming, since a large part of the energy delivered to the fluid by the impeller is discharged from it in the form of kinetic energy, and must be transferred to pressure energy in the volute. Therefore, a knowledge of the flow conditions in the volute is very desirable.

Daugherty (9) has combined a theoretical analysis with a study of the actual performance characteristics of certain pumps. One result of particular interest is his conclusion that the vane angle and the actual relative exit angle may differ by as much as five to ten degrees.

Experimental Investigations. Probably one of the first experimenters to study the flow conditions inside an actual rotating hydraulic machine was Francis (10) in 1851. In his Tremont turbine test he inserted a vane in the runner discharge, which gave the direction of the water leaving the wheel.

Photographic studies have been made by Fischer and Thoma, (8, 11) Oertli, (12) Stiess, (13) Closterhalfen, (14) and others. Fischer and Thoma worked with a pump having an open impeller and a glass side. The flow was made visible by dye injections at various points in the impeller passages. This was studied by use of a rotating prism which made the impeller appear to stand still, and was photographed by a rotating camera. Closterhalfen also used a pump with a transparent case, and in addition measured the pressures at some points along the vanes. Oertli showed that the flow in the impeller was not exactly two-dimensional.

All of these studies were carried out on pumps especially constructed for the purpose, with the design modified to permit of radial plane windows and other necessary modifications. The heads, capacities, efficiencies, and Reynolds' numbers were all low. The two latter indicated that there is good reason to expect a difference between characteristics of flow found in these pumps

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

and those existing in modern large, high-efficiency machines. Nevertheless, these studies have been very valuable in pointing out the discrepancies between the present theoretical treatments and the actual flow conditions, and also in showing the way to carry the work further.

It should be mentioned that Yendo (15) used pressure-measuring holes in the guide vanes of a turbine pump to obtain the slip coefficient. However this method does not give a measurement of the magnitude of the impeller exit velocity.

The first study of internal-flow characteristics to be undertaken in the hydraulic laboratories of the California Institute of Technology was initiated in the fall of 1931 by the present authors. The development of both instruments and technique had reached such a state by the spring of 1933 that a master's thesis was presented by Binder (16) on an investigation of the flow characteristics in the volute of one of the laboratory pumps.

In 1936, Kasai (17) reported his studies of 1933 and 1934. His instruments and method of attack followed closely those outlined in Binder's thesis (16). Although Kasai also used a specially constructed pump which had a vortex chamber between the impeller and the volute, it was of a reasonable capacity and had a good efficiency. Therefore, it is felt that his work represented a definite advance over the investigations previously discussed.

Expansion of Facilities. In the fall of 1933, the design and construction of a new hydraulic-machinery laboratory was started under a cooperative agreement between the Metropolitan Water District of Southern California and the California Institute of Technology. A description of this laboratory and its equipment has been given by Knapp (22). The program of investigation of this laboratory offered exceptional incentives to continue the work already started on the internal-flow conditions. The laboratory equipment available was adapted very well to such a study, as it offered means for exact control of all test conditions, and instruments for making precision measurements of the pump performance. A group of pumps from different manufacturers were available upon which this investigation could be carried out. They had all been selected carefully to represent the best practice in efficiency and general performance. They were of sufficient size (7 and 8 in. discharge) so that results obtained from them could be considered typical for high-head high-capacity units. In addition to having such satisfactory facilities available, it was felt that the severe conditions under which the Aqueduct pumps would operate necessitated a thorough knowledge of the internal-flow characteristics in order to insure maximum efficiency and trouble-free operation. Therefore, it was decided to proceed with the investigation which is the main subject of this article.

Objectives. The chief experimental objectives of this study have been to obtain a complete analysis of both instantaneous and average values of pressures and velocities in the volutes of the pumps investigated.

It was felt that a knowledge of the instantaneous velocities close to the impeller discharge, together with measurements of their variation with phase, i.e. with the relative position of the impeller passage, would prove very valuable in analyzing the flow in the impeller itself, since from such measurements the velocity distribution at the discharge end of the impeller passages can be calculated.

The knowledge of the average values of velocity gave promise of being useful both in aiding to understand the flow in the volute itself, and in ascertaining the changes in flow conditions in a given impeller passage during each revolution as it discharged into successively different parts of the volute.

The pressure distribution, taken together with the velocity distribution, not only should help to explain the flow characteristics, but also should furnish a basis for calculating the radial forces acting on the impeller.

With this brief discussion of the objectives of the investigation as a background, a description of the methods and instruments used will be presented before the experimental results and the conclusions are offered for consideration.

Technique of Measurement

A general discussion of the experimental methods employed to measure velocity vectors will be given first, followed by a short discussion of each major instrument. No attempt will be made in the present paper to give all of the details of the technique used, but it is hoped that a more complete description will be presented in a subsequent article.

Fig. 1 is a diagram of the apparatus used. Briefly stated, the method employed was to insert a special direction-finding pitot tube across the volute. A sampling slide valve was inserted in each of the two connections from the pitot tube to the special differential gage. These slide valves opened for a short interval

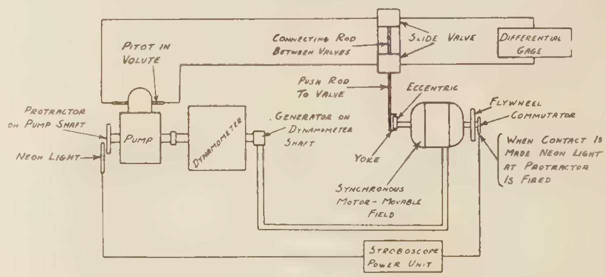


Fig. 1 Arrangement of Test Apparatus

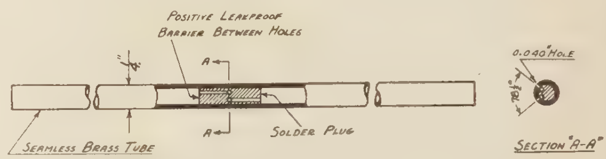


Fig. 2 Special Pitot Tube Used in the Tests
(Note: Holes drilled accurately in the same plane, normal to axis of tube.)

of time each revolution of the pump, which resulted in a series of pressure impulses to the gage. Means were provided for shifting the phase between the pump shaft and the valve opening. A stroboscope indicated the position of the opening. Thus, the velocity could be measured as any particular point of the impeller passed the pitot tube.

The Pitot-Tube Measurement. Fig. 2 shows the pitot tube used, which is sometimes called a "direction-finding" pitot by wind-tunnel experimenters. Fig. 3 shows the method of insertion in the different pump volutes.

In any flow measurement it is not difficult to determine the true total head (velocity plus static), for the total head is obtained by placing an opening normal to the stream. However, for an accurate determination of velocity, it is also necessary to have a precise measurement of the static head, and this is much more difficult to secure. One of the main features of this pitot tube is that it gives an accurate measurement of static head in turbulent flow.

Considering the pressure distribution around a small cylinder across a stream, it is known that there is a critical angle with the direction of flow at which the velocity pressure has no effect. This means that, having an opening at the critical angle with the

flow direction, the pressure transmitted to a gage will be truly static and unaffected by any influence of velocity.

Dryden (18) and Fechheimer (19) were the early contributors toward the development of this type of pitot tube for air measurements. Fechheimer found the critical angle to be $39^1/_4$ deg. The authors have checked this critical angle and the construction of $^1/_4$-in. and $^3/_{16}$-in. diameter pitot tubes by observing the position of the hole in a stream of known direction where the static pressure was known. This check gave an angle of $39^1/_4$ deg for the velocity range met in these pumps tests. It is interesting to note that these later measurements were made in water, but that the Reynolds number was substantially the same as that used by Fechheimer.

Referring to Fig. 2, the small pressure openings were possible because of the use of a special differential gage to be described later. In using the tube in the pump volute it was necessary to "balance" the tube. Each static hole was connected to one side of the differential gage. The pitot tube in the unknown stream was rotated about its own axis until the pressures at each hole were the same, in other words, the differential pressure was zero. At this position velocity pressure had no effect on either hole, and

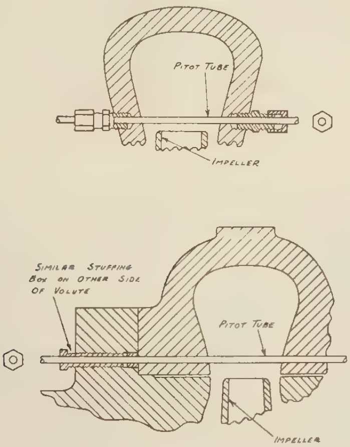


Fig. 3 Methods of Inserting Pitot Tube in Different Pump Volutes

(Top: Radial section through volute of Byron Jackson double suction pump. Bottom: Radial section through volute of single-suction Worthington pump.)

either hole could be used to measure static pressure. The bisector of the angle between the holes gave the direction of flow. The dynamic pressure was then obtained by placing one opening normal to the direction of flow, i.e., by simply rotating one hole back into the stream $39^1/_4$ deg. Thus, with the values of the directly measured total and static heads, the difference gave the velocity head, and the measured angle gave the direction of the velocity vector.

An interesting and delicate technique was finally developed for constructing the pitot tube. After carefully tinning and then cleaning the inside of the brass tube, the holes were accurately drilled in a special jig. A piece of clean polished piano wire was inserted through one hole from the inside and extended out one end, while another piece of piano wire was arranged likewise for the other hole. A small metal plug was placed about $^1/_2$ in. below the plane of the holes and small pieces of solder filled in; heating carefully in an electric heater (for close temperature control) and in a reducing atmosphere (to prevent foreign matter from interfering with the perfect barrier) the solder plug, or barrier, was formed between the holes. Pulling out the piano wire as the solder solidified left the desired passages. The tube was then tested for the barrier, and checked for angle and velocity accuracy.

Special apparatus was devised for checking these pitot tubes. Using a free jet, many tests were made on the magnitude of velocity measurement, and very close agreements obtained for the range of high velocities met in pump traverses. Using this special apparatus with a closed jet, a wall correction was found which was applied to pump traverses. This wall-correction curve is shown in Fig. 4.

Special Differential Gage. An ordinary mercury or water U-tube manometer would be out of the question on these fluctuating pressure measurements. To obtain a reading in a system using an ordinary U tube, an appreciable flow is required through the pitot pressure openings and the connecting leads; hence, a

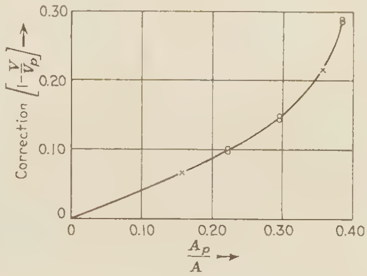


Fig. 4 Pitot-Tube Wall Correction

(A_p = projected area of the pitot tube, A = area of pipe without pitot tube, V_p = mean velocity indicated by pitot tube traverse, and V = velocity = quantity per unit time divided by A.)

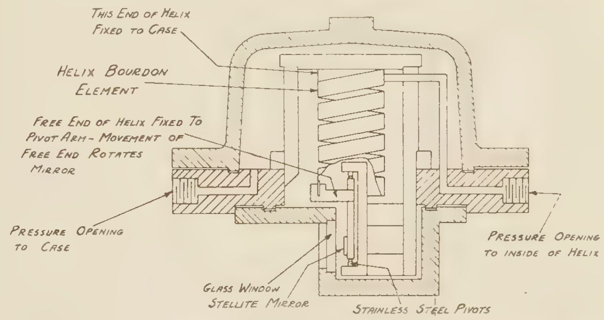


Fig. 5 Special Differential Gage for Fluctuating Pressure Measurements

reading could not be obtained in a reasonable time. This difficulty was overcome by the development of a special differential gage. Its main features are that only a very small amount of flow is required for operation, the gage is very rapid, sensitive, and accurate. Many experimenters using pitot tubes have been limited by the use of U-tube manometers. Such a manometer requires that the pitot pressure openings be large enough to avoid excessive damping, while this special differential gage permits the use of much smaller pressure openings, and therefore smaller tube diameters.

Fig. 5 shows the internal construction of the differential gage. Since the helix element is the same as is used on pressure-recording instruments, this differential gage can be adapted to any desired accuracy and range of pressure by a suitable selection of helix. One end of the helix element is fixed, while the other end is free to move. The free end is so connected as to cause a rotation of the stellite mirror when the free end moves. Water pressure is applied to both the inside and the outside of the Bourdon element, the whole mechanism being in water in a closed case. Thus, when the differential pressure changes, the free end of the Bourdon element rotates the mirror. The mirror arrangement magnifies this movement with the aid of an optical system. A light source sends a beam of light through the glass window to

strike the mirror. The reflected ray is focused on a graduated scale. The complete gage setup is shown in Fig. 6.

The gage was calibrated with a deadweight gage tester, and gave a straight-line calibration curve. Repeated tests over long periods of time have shown that this gage holds its calibration precisely. Tests have shown that the scale deflection depends solely on the differential pressure and is independent of the absolute pressure.

One interesting feature of this gage is that it requires no time to give a pressure reading. In this gage there is no appreciable flow of water, it is practically a constant-volume system.

Fig. 6 Setup of the Differential Gage

Sampling Valve and Phase Shifter. With the pitot tube and the special differential gage, measurements of average velocity can be made in the pump volute. This in itself is an improved technique, but further developments were made. Because of the extremely minute flow required to operate the differential gage it was possible to use a slide valve to sample the pressure transmitted from the pitot as any particular point on the impeller passed the pitot tube. The following description will show how this was accomplished.

Referring to Fig. 1, the generator on the dynamometer shaft drove the synchronous motor at one-half pump speed. An eccentric on the motor shaft worked in a yoke to impart simple harmonic motion to the push rod driving the two valves. Each valve opened twice (back and forth) for every revolution of the synchronous motor, which meant one valve opening per revolution of the pump.

When the valves opened, the commutator contact would fire the neon light at the protractor on the pump shaft (one firing per one revolution of pump). The bolts between the field and the end bells of the valve motor were removed, and means provided for rotating the field of the motor. Thus there was a positive mechanical-electrical connection between the pump shaft and the slide valve, and by simply rotating the field of the motor it was possible to change the phase relation between the pump shaft and the time of opening of the valve. It was possible to change the phase by 360 deg, while the stroboscope always gave a precise indication of the position of opening of the valve.

Fig. 7 shows the construction of one of the valves. E. R. Lockhart (20) helped work out the details of the sampling valves. The valves are duplicates, and accurately positioned to open at exactly the same time.

Use was made of the fact that the eccentric and yoke imparted simple harmonic motion to the valve, with the result that at the middle of the valve travel the velocity is a maximum while the acceleration is zero. The valve therefore was set to open at the middle of the travel. With a slot thickness of 0.005 in., the time of opening corresponds to an angular rotation of 5 deg of impeller. Tests have shown that the motion of the valve at any speed has no effect on the pressure transmitted through the valve. This is probably due to the fact that the valve opens at the point of zero acceleration. Whether or not this is the complete explanation, it is an experimental fact that the slide valve has no effect on the pressure transmitted.

One very interesting check was made. With the apparatus installed on a pump, in place of the pitot tube an oscillating pressure of known frequency was applied to the slide valve. The pump was run at various speeds, each different from the known frequency of the applied pressure. For each case the number of "beats" per minute, as shown by the differential gage, corresponded exactly to the difference between the pump speed and the cycles per minute of the applied pressure.

Tests have shown that each slide valve when closed does not leak. A typical installation is shown in Fig. 8.

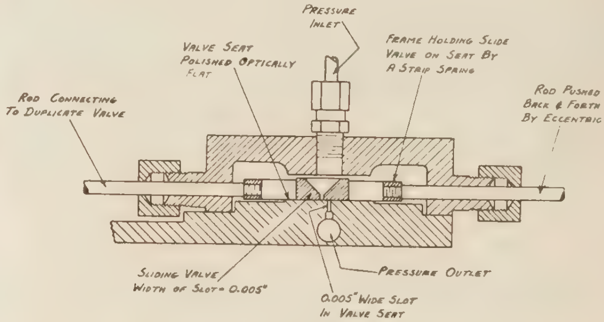


Fig. 7 Construction of Slide Valve

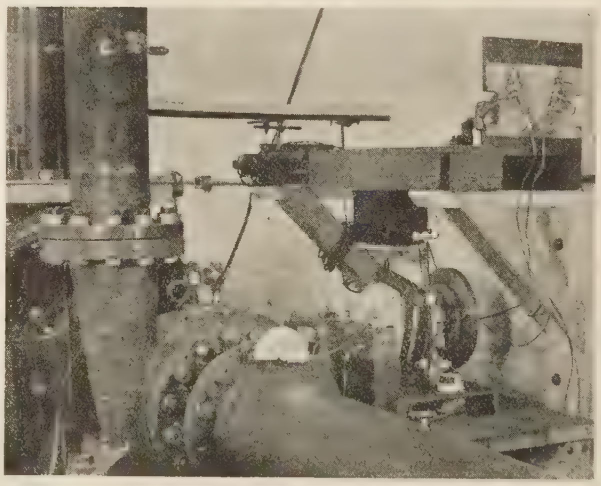


Fig. 8 A Test Installation

General Remarks on Technique of Measurement. This technique of instantaneous velocity measurement is possible because the system from pitot tube to gage has practically no volume changes; the pressure is transmitted by the water without any appreciable flow. Pressure waves in the connecting leads might cause trouble, but the length of the leads was reduced to a minimum by placing the slide valve as close as possible to the pump.

Presentation of Test Results

Notation Used in Measurements. For designating valve opening, one vane-tip edge was chosen as a zero reference. If the valve opened as the zero reference mark passed the pitot tube the "phase angle was 0 deg." If the valve opened as some other point on the impeller passed the pitot tube, this point was referred to the zero mark as so many "degrees phase angle," where this

angle is measured in the opposite sense to that of the pump rotation, i.e., the point lags the zero reference.

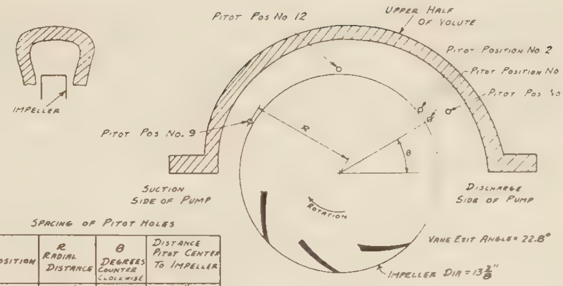
For designating the axial position of measurement across the volute, "center" means over the center of the impeller, while "right" or "left" refers to the side from this center. On the double-suction pump "right" and "left" were used with the observer facing the pump suction flange. On the single-suction pump the "right" side refers to the suction side of the pump. The test results and curves from both pumps will be given first, to be followed by a combined discussion.

Instantaneous Velocity Measurements on Byron Jackson 8-In. Double-Suction Centrifugal Pump. The pump rating and dimensions are as follows:

Capacity = 2400 gpm
Total head = 360 ft
Speed = 2500 rpm
Specific speed = 1400
Impeller outside diameter = $13^3/_8$ in.
Impeller inside width = $1^{12}/_{32}$ to $1^{13}/_{32}$ in.
Impeller outside width = $1^3/_4$ in.
Number of vanes = 8

All tests on this pump were made at 2000 rpm. The maximum efficiency at 2000 rpm was 84.6 per cent, which was the same as at the rated speed. All tests were made at plus 40 ft inlet head. Hydraulically, this pump was better than appears here. Before these pitot-tube measurements were made, this pump had received some severe treatment in previous tests, with the results that the leakage losses were increased as the efficiency decreased from an original value of 85.8 per cent to 84.6 per cent.

Fig. 9 shows the spacing of the pitot-tube stations in the volute. Since this was a horizontally split-case double-suction pump, it was not possible to provide pitot stations in the lower half of the volute, and therefore no measurements could be made in the vicinity of the tongue.



Position	R Radial Distance	θ Degrees Counter Clockwise	Distance Pitot Center To Impeller
1	7 1/16"	30.3	3/8"
2	6 15/16"	37.8	1/4"
6	8 1/16"	29.3	1 3/4"
9	7 1/16"	149.7	3/8"
12	7 1/16"	90.0	3/8"

Fig. 9 Spacing of the Pitot-Tube Stations in the Volute of the Byron Jackson Pump

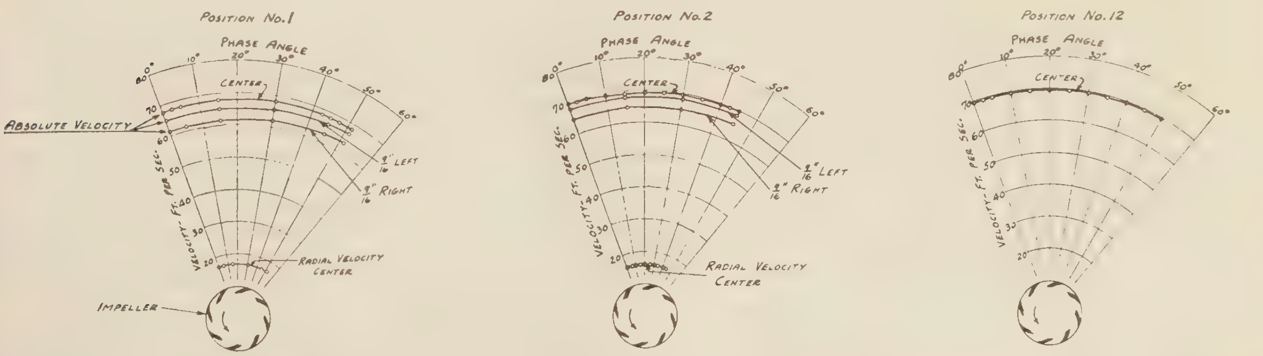


Fig. 10 Instantaneous Velocity Distribution Between Vanes in the Bryon Jackson Pump
(Pump speed = 2000 rpm, and normal capacity = 4.65 cfs.)

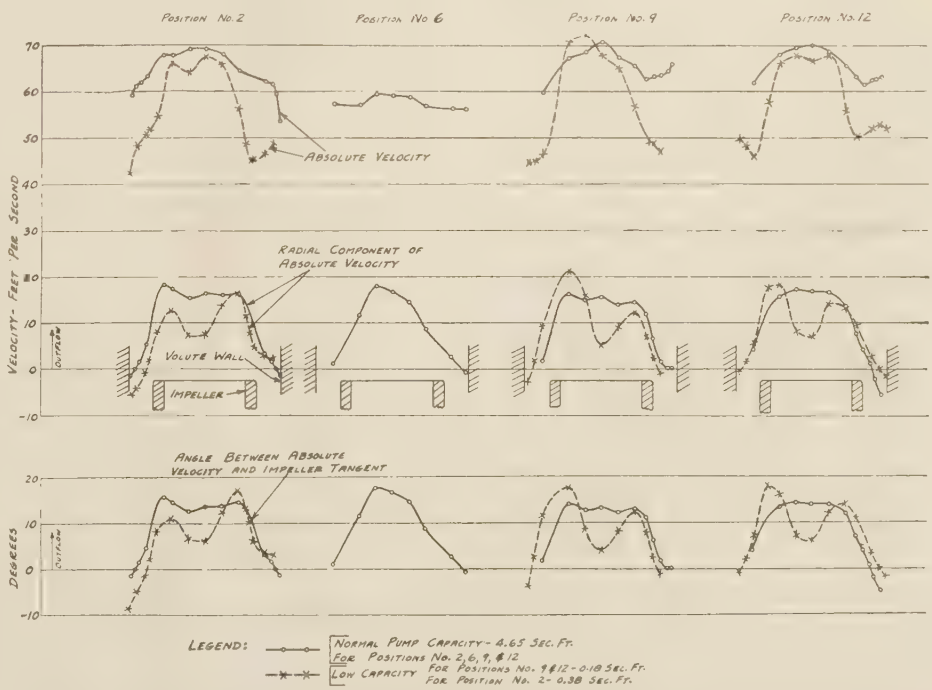


Fig. 11 Instantaneous Velocity Traverse Across the Volute of the Byron Jackson Pump
(Pump speed = 2000 rpm, impeller velocity = 116.8 fps, and phase angle = 0 deg.)

Fig. 10 shows the results of measurements to find the velocity distribution between vanes. For each set of measurements, the pitot tube was kept at a fixed position across the volute, while the phase angle was varied to traverse the impeller passage (by rotating the field of the synchronous motor driving the slide valves).

Fig. 11 shows the profiles obtained from pitot-tube traverses across the volute, each traverse being made at a constant phase angle.

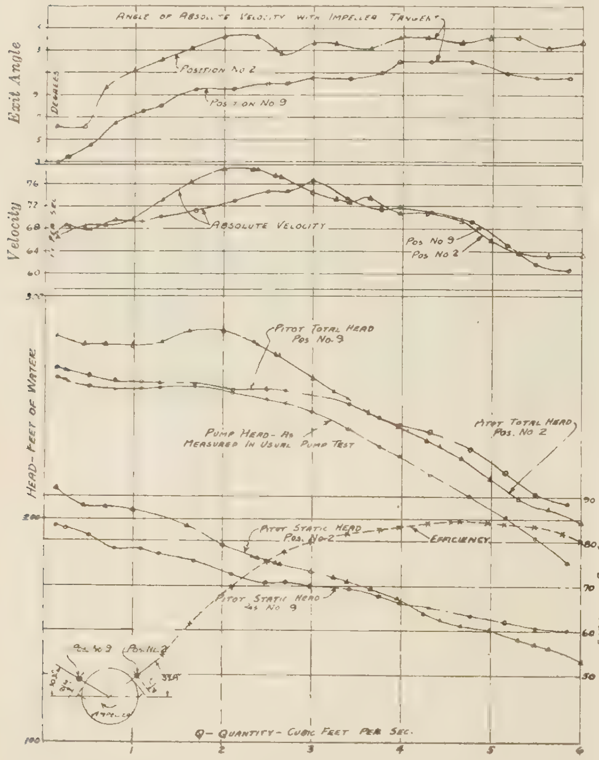


Fig. 12 Pitot-Tube Measurements at Different Discharges of the Byron Jackson Pump

Fig. 12 shows the pitot-tube measurements at different pump discharges. With the pitot at one position in the center of the volute, the pump capacity was varied.

Instantaneous Velocity Measurements on Worthington 7-In. Single-Suction Centrifugal Pump. The pump rating and dimensions are as follows:

Capacity = 2400 gpm
Head = 360 ft
Speed = 2900 rpm
Specific speed = 1720
Impeller outside diameter = $12^1/_2$ in.
Impeller inside width = $1^7/_{32}$ to $1^{15}/_{64}$ in.
Impeller outside width = $1^{17}/_{32}$ in.
Number of vanes = 7

All tests on this pump were made at 2500 rpm. The maximum efficiency at 2500 rpm was 88.6 per cent which was the same as at the rated speed. All tests were made at plus 40 ft inlet head.

Extensive measurements were made at "normal," "low," and "high" pump discharges. Normal pump discharge is that at the point of maximum pump efficiency. Low refers to a discharge of about 19 per cent of normal, while high refers to a discharge of about 142 per cent of normal.

Fig. 13 shows the spacing of the pitot-tube stations in the volute. It should be noted that there are two points of difference in the location of these stations as compared to those of Fig. 9. First, they are spaced completely around the volute, and second, they are all located at a constant radial distance from the impeller.

Fig. 14 shows the results of measurements as the pitot tube was kept at a fixed position across the volute and the phase angle varied.

Figs. 15, 16, and 17 show the profiles obtained from traverses across the volute, each traverse being made at a constant phase angle, and each figure referring to a different pump capacity. The traverses at high capacity are not complete, but the measurements are useful to some extent in a comparison with the results of tests at normal and low capacities.

Fig. 18 shows, for each pump capacity, a plot of average radial velocity vs. angle around the volute. Each point represents the average value of the corresponding profile found in Figs. 15, 16, or 17.

Fig. 19 shows the static-pressure distribution around the volute as given by the pitot tube. The static pressure plotted is that developed by the pump, and thus does not include the inlet pressure. The dashed horizontal lines indicate the mean static-pressure value for each curve.

Fig. 20 shows both the unbalanced static pressure and the momentum forces acting on the impeller. In the absence of other definite information the outside outlet width of the impeller was taken as the area over which the static pressure acts.

Fig. 21 shows the average direction of the relative exit-velocity vectors at the different pitot stations. From each traverse across the impeller width, the average radial velocity and the average tangential velocity were computed. From these two averages and the impeller peripheral velocity a relative exit angle *B* was calculated for each traverse.

Dimensions of Pitot Locations

Position No.	"R" Radial Distance	"θ" Degrees	Distance Pitot Center To Impeller
A	6 70"	25.5	0 45"
B	6.68"	49.8	0.43"
C	6.68"	67.1	0.43"
D	6.66"	90.2	0.41"
E	6.67"	135.4	0.42"
F	6.68'	157.6	0.43"
G	6.69'	224.9	0.44'
H	6.68"	269 1	0.43"
I	6.72"	295.4	0.47"
J	6 71"	314 5	0.44'
K	6.71"	334 1	0.46
L	6.71	354 8	0.46"

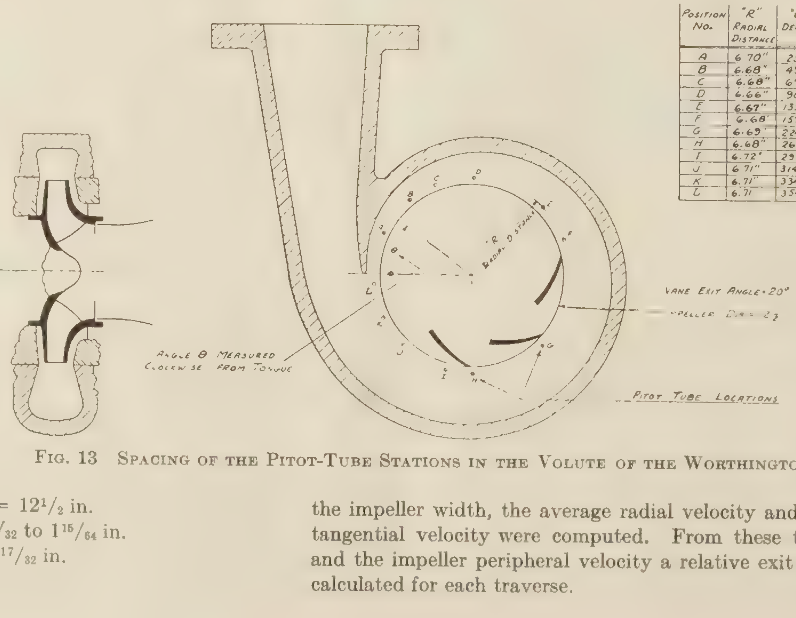


Fig. 13 Spacing of the Pitot-Tube Stations in the Volute of the Worthington Pump

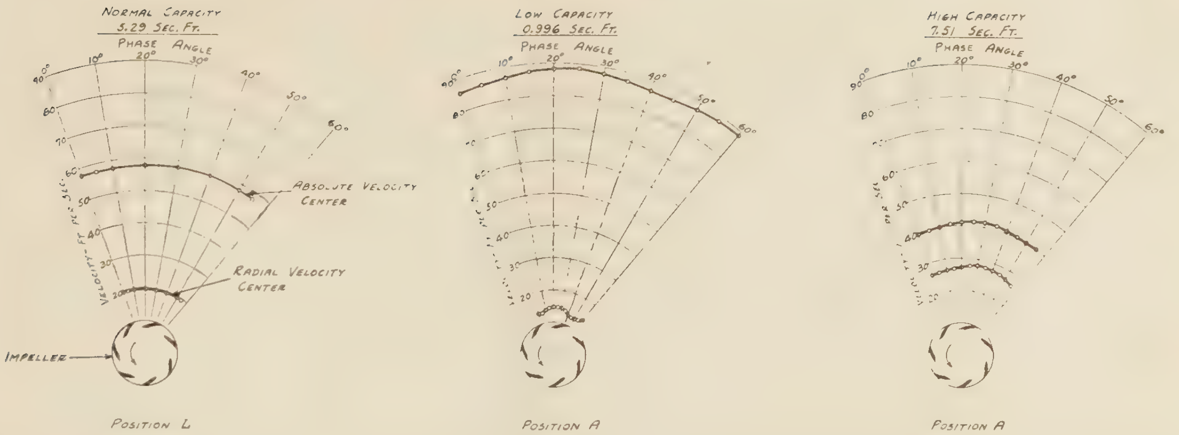


Fig. 14 Instantaneous Velocity Distribution Between the Vanes of the Worthington Single-Suction Pump at 2500 Rpm

Discussion of Results

It has been seen in the description of the test results that the measurements taken on the double-suction pump are not as complete as those made on the single-suction pump. However, in so far as they do duplicate each other, it is instructive to consider the data from both pumps simultaneously. Before doing so it should be noted that the double-suction pump has the lowest specific speed, even when both calculations are made on the same basis. If the specific speed of the double-suction machine is calculated on the basis of one half the rated capacity, as is often the practice, then it becomes even lower, i.e., it reduces to about 990, compared to 1720 for the single-suction pump.

Instantaneous Velocity Distribution in Impeller Passages. Although no measurements could be made in the impeller passages themselves, much could be inferred concerning the velocity distribution at the discharge of the impeller passages from the instantaneous measurements taken with the pitot inserted in the volute very close to the impeller. As the phase angle at which the slide valve opened was shifted, the velocity gradient between the vanes was measured. The results obtained are seen for the two pumps in Figs. 10 and 14. The most striking thing to be noticed is that there is very little velocity variation observable between the vanes at normal pump discharge. This is not in agreement with the normal conception of the impeller flow, in which it is often assumed that there is a dead water space or even a backflow along the low-pressure side of the vane. Before any conclusions are reached it is necessary to consider several characteristics of the measurements:

(*a*) The velocity distribution here obtained is not a true instantaneous picture of the flow from the volute but is rather the time variation of the velocity as the impeller passage passes a given station. To obtain the actual instantaneous velocity distribution it would have been necessary to have had a series of pitot stations spaced a few degrees apart around a portion of the impeller periphery, and to have taken a measurement with the correct phase angle at each of the stations.

(*b*) The slide valve is opened an appreciable time, i.e., about 5 deg of arc. Therefore an individual measurement is an average and not an instantaneous value, and, due to wire drawing, it is not an arithmetical average.

(*c*) Due to structural features of the pumps, clearances of from $^1/_8$ to $^5/_{16}$ in. between the impeller and the pitot were necessary. Some change of velocity could therefore occur between the points of discharge and measurement. For example, this together with (*b*) explains why the flow is not zero during the time the vane itself is passing the measuring point.

After taking all of the foregoing factors into consideration, the conclusion is still unavoidable that in high-efficiency pumps, operating under conditions of normal discharge, the velocity distribution across the impeller passage discharge is surprisingly uniform.

Fig. 14 shows that for capacities either above or below normal, some velocity gradients are observable. However, in no case are they as great as previous studies have indicated.

Velocity Profiles Across the Volute. Figs. 11, 15, 16, and 17 show the velocity profiles obtained by making traverses across the volutes from wall to wall at the different stations. It will be noted immediately that there are no marked breaks in the profiles to show the locations of the impeller shrouds. In this connection no attempt should be made to find zero flow between the shroud and the case at any single pitot station. Although the total flow across this space must be equal to the leakage through the wearing rings, it is very possible to have flow into the space in one region of the volute and out in other regions. In fact, this circulation can act as an energy pump by entering this space from the volute at a relatively low velocity and later returning to another section of the volute with a higher velocity. For example, in Fig. 15 the radial-velocity components show a net inflow at relatively low velocity to this space from stations *A* to *G* and outflow at higher velocity from station *H* to *L*. It will be remembered that originally pump volutes were built with small clearances between the walls and the impeller periphery, but that efficiencies were improved when the clearances were made much larger. The energy flow previously mentioned may account for some of this improvement, because, with the close clearances, the energy imparted to the fluid in these spaces by disk friction on the shrouds is trapped in the spaces and must be dissipated without benefit to the pump performance, while with ample clearances at least a part of this energy may be carried out into the volute and utilized.

An inspection of the radial-velocity distributions for the low-capacity readings in Fig. 11 shows that under this condition the flow has two high-velocity peaks at each station. An obvious suggestion is that this is due to the double-suction impeller, which is fundamentally two impellers placed back to back. However, this is immediately seen to be erroneous when the same peaks are found in Fig. 16, which is plotted from measurements of the single-suction pump. The most reasonable explanation of these peaks appears to arise from a consideration of the centrifuge action of the shroud. Professor von Kármán has suggested a calculation to help explain this matter. In the following calculation no claim is made to express exactly the

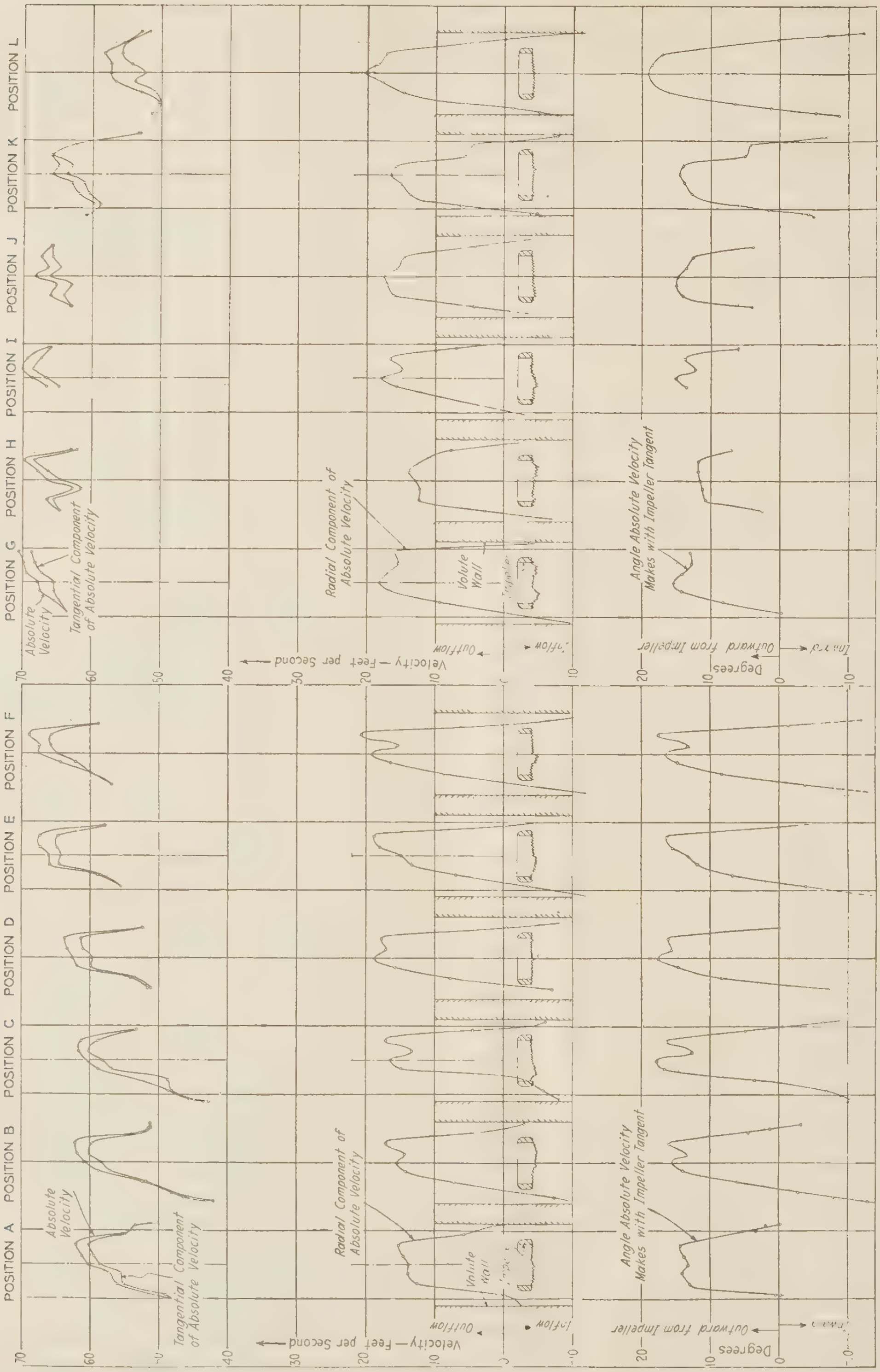


FIG. 15 VELOCITY TRAVERSES ACROSS VOLUTE IN THE WORTHINGTON PUMP

(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 274 ft, impeller peripheral velocity = 136.3 fps, and normal capacity = 5.29 cfs.)

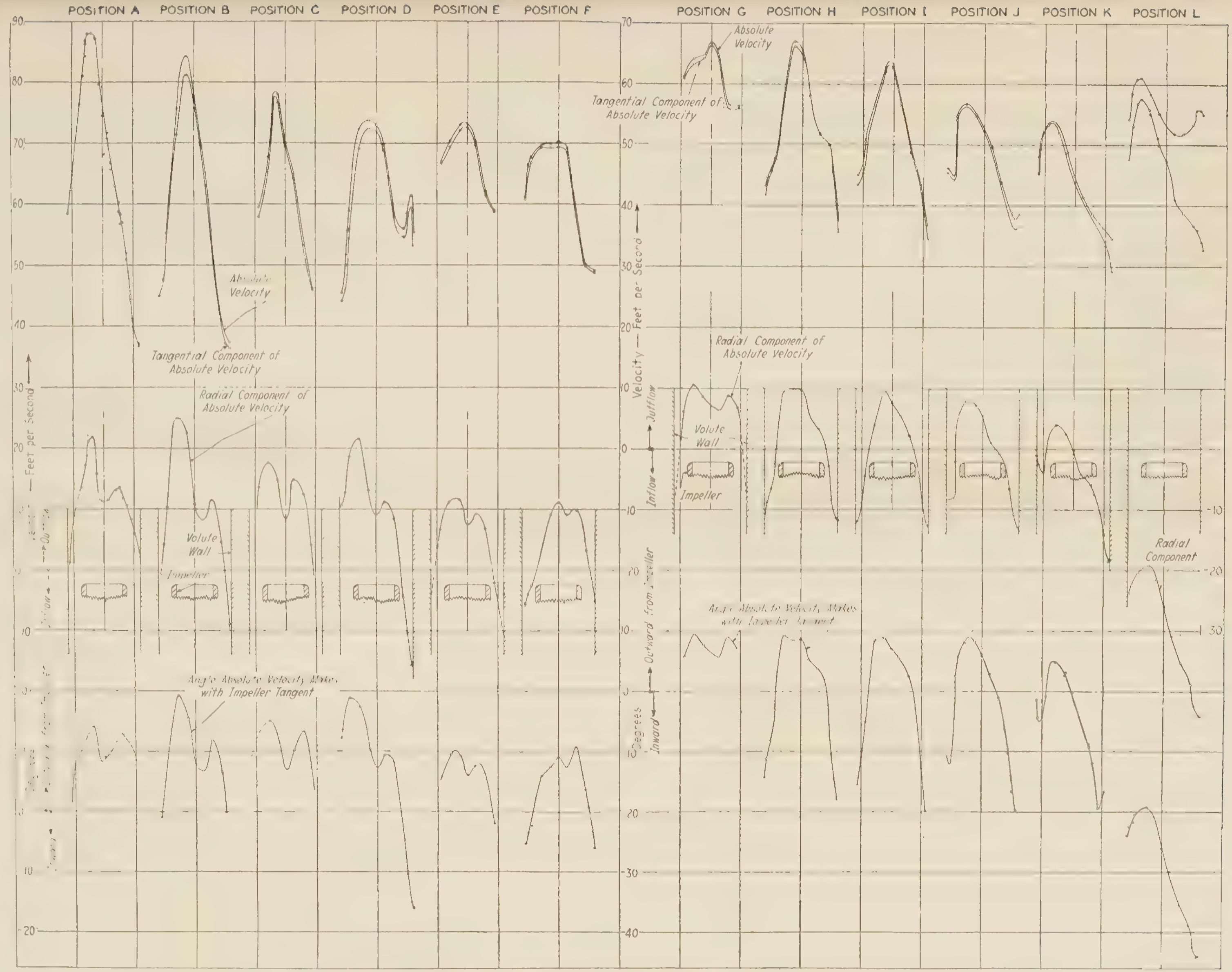


Fig. 16 Velocity Traverses Across Volute in the Worthington Pump for Low Capacity

(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 336 ft, impeller peripheral velocity = 136.3 fps, and low capacity = 0.996 cfs. Note change of scale between positions F and G.)

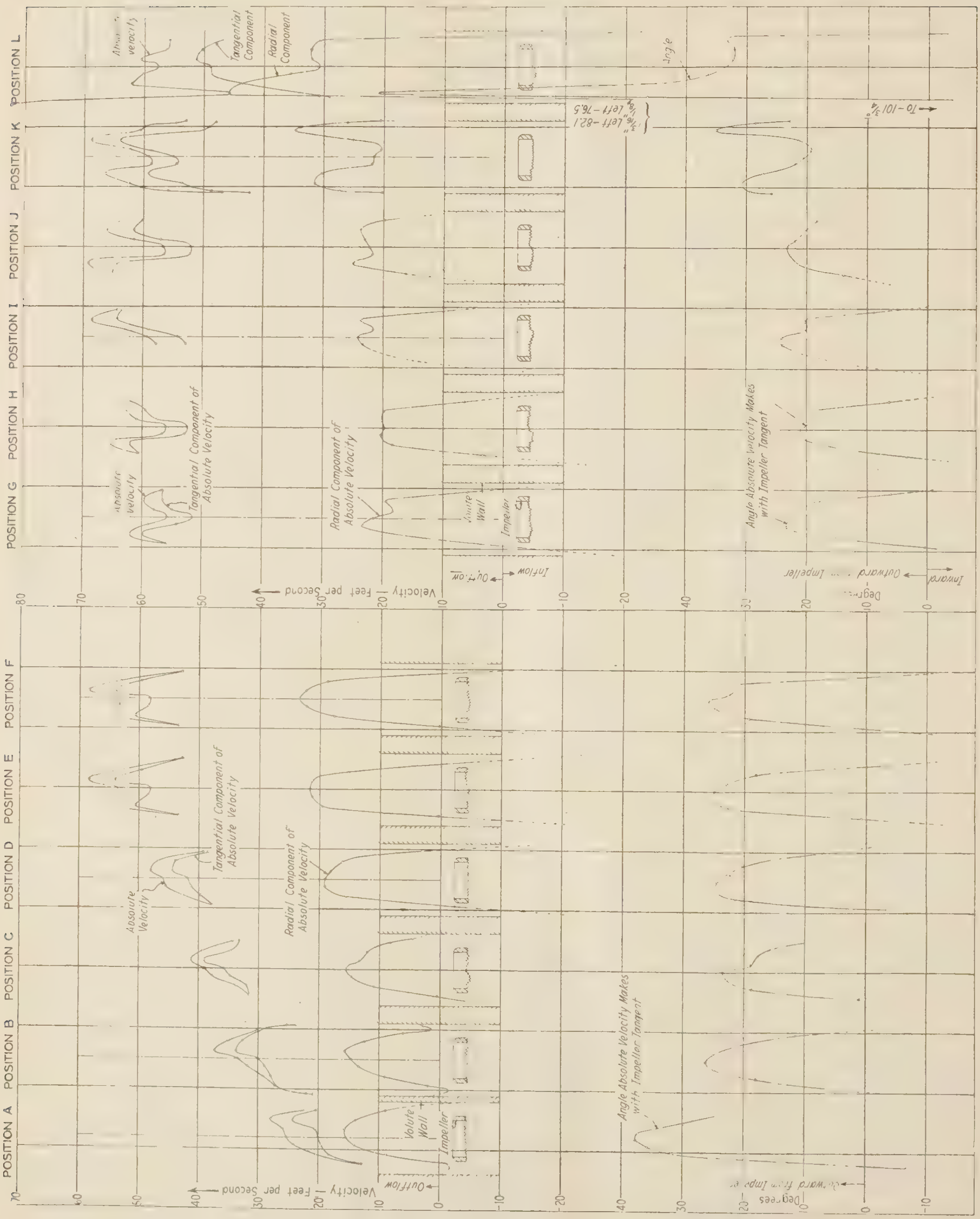


Fig. 17 Velocity Traverses Across Volute in the Worthington Pump for High Capacity

(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 134 ft, impeller peripheral velocity = 136.3 fps, and high capacity = 7.54 cfs. Note change of scale between positions F and G.)

complicated conditions in a pump, but the computation serves to give the order of magnitude of the peaks. Professor von Kármán (21) has treated the problem of the frictional resistance of a rotating disk for the case of turbulent flow. He considered a smooth flat disk wetted on one side. The various momentum changes were taken into account, and the velocity distribution in

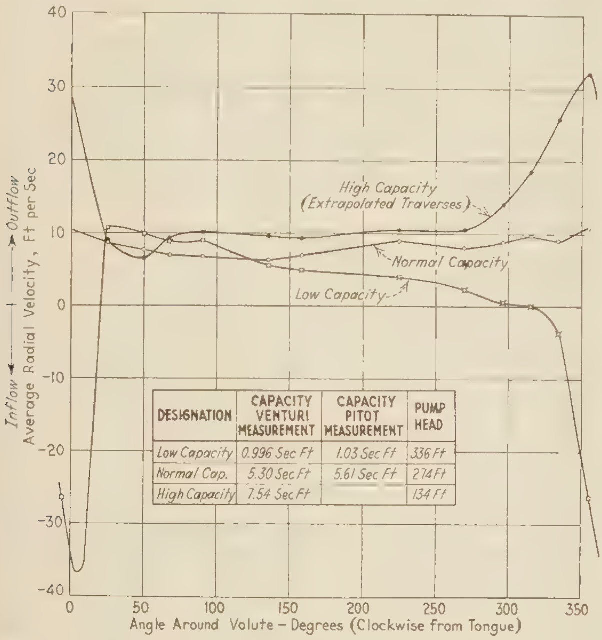


Fig. 18 Average Radial Velocity at Different Angles Around the Volute of the Worthington Pump
(Pump speed = 2500 rpm, and impeller peripheral velocity = 136.3 fps.)

the boundary layer was assumed to follow the seventh-root law. The following expressions were derived in the treatment

$$C_o = 0.162\ R\omega$$

$$\delta = 0.522\ R\left(\frac{\nu}{R^2\omega}\right)^{1/5}$$

where C_o = maximum radial velocity in the boundary layer, R = radius of disk, ω = angular velocity of the disk, δ = thickness of the boundary layer, and ν = kinematic viscosity of the fluid. Applying the two relations to the two pumps under consideration, and taking for R the radii of the peripheries of the impellers, the following values are obtained: For the double-suction pump C_o = 18.9 fps, and δ = 0.15 in. For the single-suction pump C_o = 22.1 fps, and δ = 0.14 in.

Figs. 11 and 16 show that some of the measured radial-velocity peaks are close to these computed values of C_o. Thus it is indicated that each shroud acts as a centrifuge to discharge a sheet of water into the volute.

The question might be raised as to why there is a lack of a "shadow" in the velocity profiles above the shroud. In both pumps the shrouds are about $^3/_{16}$ in. thick, and in this space there should be no radial flow. However, it is quite possible that in the short radial distance (between the impeller periphery and the measuring tube) the flows from both sides could diverge, and that these divergencies could combine to give an appreciable positive velocity over the shroud thickness.

The difference in the velocity of these two flows, together with their initial separation due to the shroud thickness, offers a possible explanation of the shift of the velocity peaks away from the computed boundary layer and toward the center of the impeller. Referring again to Figs. 15, 16, and 17, it will be seen that there are unsymmetrical peaks on the absolute-velocity profiles. For both normal and high capacities, these peaks are on the suction side of the impeller center, while for low capacity the peaks are on the shaft side. It would be interesting to observe whether or not this shift of the position of the peaks occurred at the same time as the shift in the direction of the thrust commonly observed in single-suction pumps.

Pitot-Tube Measurements for Variable Pump Capacity. Fig. 12 shows the comparison between the head developed by the double-suction pump and the corresponding static and total heads as measured at two stations in the volute, for a wide range of capacities. Although the measurements were taken with the pitot fixed at the center line of the impeller, the static-head readings

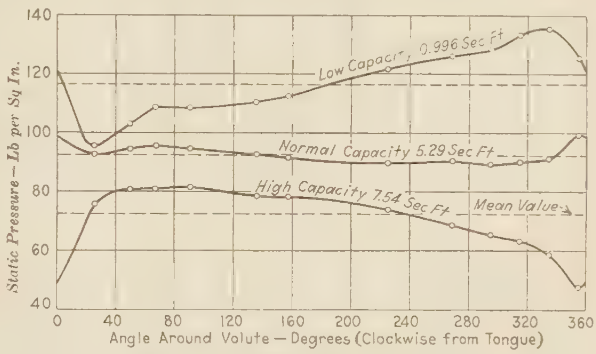


Fig. 19 Static-Pressure Distribution Around the Volute of the Worthington Pump at 2500 Rpm

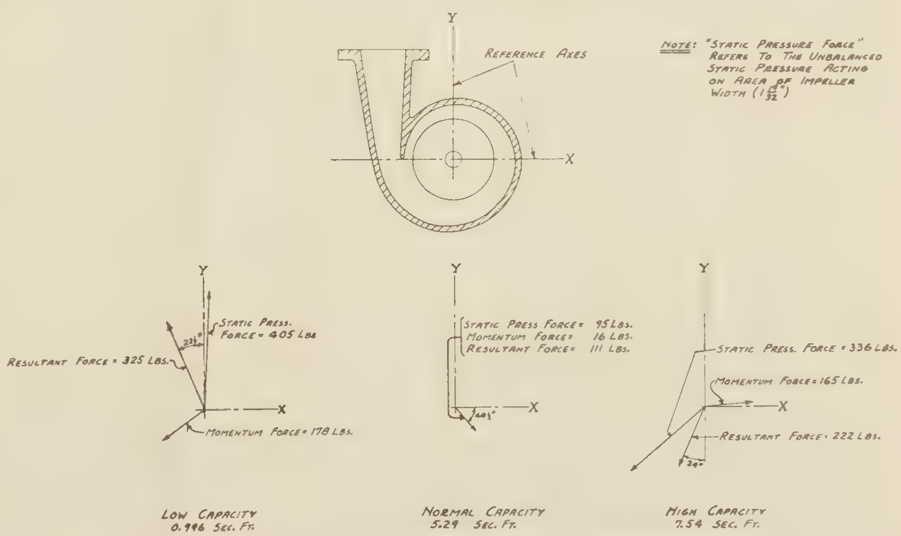


Fig. 20 Unbalanced Radial Forces Acting on the Impeller of the Worthington Pump

probably represent average values across the width of the volute, since it was found that the static pressure was relatively constant throughout the traverse. On the other hand, the total-head readings are not so representative, as shown by the absolute-velocity traverses of Fig. 11. The difference between the static pressures at the stations shows that there may be a possibility of an unbalanced radial force on the impeller, especially in the low-capacity region. Since the two curves cross in the vicinity of normal capacity, it might be expected that the direction of the unbalanced force would reverse in the high-capacity region.

Radial-Velocity Distribution Around the Volute. The complete ring of pitot stations provided in the single-suction pump has made it possible for the first time to secure sufficient data to compute the radial-velocity distribution around the entire volute. Fig. 18 shows the three distribution curves obtained.

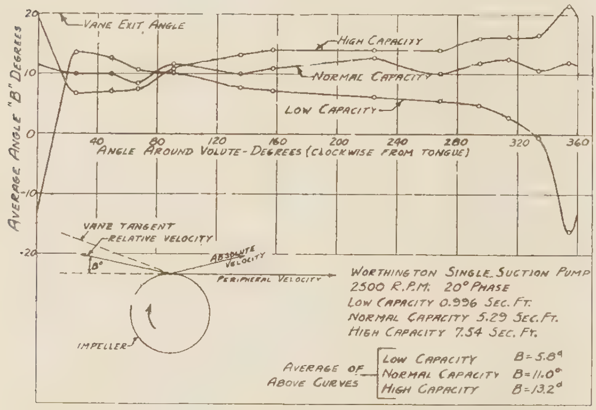


FIG. 21 DIRECTION OF AVERAGE RELATIVE EXIT VELOCITY AT DIFFERENT PITOT STATIONS AROUND THE VOLUTE OF THE WORTHINGTON PUMP

The first item to note is that the three capacities show markedly different characteristics. For normal capacity, the radial velocity is relatively constant around the entire volute, although it is by no means exactly so. At high capacity there is a region of uniform velocity extending over about 200 deg. The remainder of the circumference, which is in the vicinity of the tongue, shows a much higher value of outflow. At low capacity there is no region of uniform flow. However in the region of the tongue there is a very striking zone of high inflow.

Both high- and low-capacity distributions lead to the same conclusion that, except for normal-capacity operation, there is always a large velocity variation in the impeller passage each revolution. Under some low-capacity conditions this becomes an actual reversal of flow. Probably this velocity fluctuation accounts for some loss in pump efficiency.

Unbalanced Radial Forces. In the operation of high-head pumps, trouble sometimes arises with the wearing rings, caused by large shaft deflections. This results in metallic contact excessive wear and, hence, increased leakage. In severe cases, shafts have been known to break due to fatigue. Fig. 19 is a plot of the static-pressure distribution around the impeller of the single-suction pump, in an attempt to study this unbalanced force. Lack of uniformity of static pressure would of course give rise to such a radial resultant force. Note that at normal capacity the static pressure is quite uniform, while very wide variations are present for both low and high capacities.

If the radial forces represented in Fig. 19 are added vectorially, the magnitude and direction of the unbalanced resultant is obtained. Reference to Fig. 20 shows that the directions of the resultants are quite different for high- and low-capacity operation. This could have been predicated from the difference in shapes of the corresponding pressure distributions of Fig. 19.

This force resulting from the unbalanced static-pressure distribution is, however, not the only radial force acting on the impeller. A nonuniform velocity distribution will give rise to an unbalanced momentum force in the same manner. This must be added to the static-pressure force to obtain the total hydraulic reaction.

The analysis of the three operating conditions presented in Fig. 20 shows that at low capacity the deflection may be about three times that at normal capacity, while for high capacity it may be twice that at normal. The latter is not so serious, since operation at high capacity is not always necessary. On the other hand, the low-capacity range is always passed through when the machine is started and stopped, and continuous operation in this region is not uncommon.

Measurements of the deflections at the impeller wearing ring during operation under the various indicated capacities have shown that the impeller movement was in a direction which agrees with that of the resultant force vectors as determined in Fig. 20. A study of the stress-strain conditions in the shaft agreed in magnitude with those calculated from the hydraulic reaction, although in general the former are somewhat higher. However, this stress-strain analysis involves considerable difficulty because of the uncertainty as to the amount of the bearing and casing deflections.

Direction of Relative Exit Velocity From Impeller. A question of considerable interest is the relation between the vane exit angle and the direction of the relative velocity of the fluid leaving the impeller. One item to note is that the relative exit angle is less than the vane angle for all points except one. Again, at normal capacity the conditions are relatively uniform around the entire perpihery, while at low and high capacity there is a wide deviation between the different stations. The averages given in Fig. 21 show that at low capacity the deviation between the vane angle and the relative velocity is 14.2 deg, for normal capacity 9 deg, and for high capacity 6.8 deg. Note that in the vicinity of the tongue the deviation reaches as much as 36 deg.

SUMMARY OF RESULTS

1 There is practically no instantaneous velocity variation in the impeller discharge at normal capacity and only slightly more at low or high capacities during the time of passage of one vane space past a measuring station.

2 There is a strong circulation between the volute and the impeller clearance space which apparently acts as an energy pump and helps to minimize losses.

3 For low-capacity conditions, double peak-velocity profiles were found, probably due to a centrifuge action of the shroud.

4 For normal capacity, the radial-velocity distribution around the volute is relatively uniform, while for high or low capacities large variations are found.

5 For low capacity, a high ratio of inflow is observed in the region of the tongue, while for high capacity a high outflow occurs in the same area.

6 Nonuniform velocity and static-pressure distribution combine to produce unbalanced radial forces on the impeller. Maximum values exist during low-capacity operation, while the forces are at a minimum for normal rates of discharge.

7 A considerable variation in the deviation between the vane exit angle and the relative velocity was observed. The average deviation was greatest for low-capacity conditions and least for high-capacity conditions. At normal discharges it was 9 deg.

ACKNOWLEDGMENT

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The Hydraulic-Machinery Laboratory at the California Institute of Technology

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This paper gives a description of the arrangement, equipment, and instrumentation of the hydraulic-machinery laboratory at the California Institute of Technology. This laboratory was designed essentially to work with problems involving high heads, high speeds, and moderately large powers and rates of flow. The instruments and equipment permit both speed and precision in testing, an overall accuracy of 0.1 per cent being attainable. For the past two years the laboratory has been used for the study of pumping problems of the Colorado River aqueduct, a project which will have pumps totaling 350,000 hp when completed.

THE Hydraulic-Machinery Laboratory is a joint enterprise of the California Institute of Technology and the Metropolitan Water District of Southern California. In November, 1933, the formation of the laboratory was authorized by F. E. Weymouth, general manager and chief engineer of the Metropolitan Water District, a three-year agreement was signed by the two organizations, and the design of the laboratory was begun. Construction was completed by the end of August, 1934, and since that time this laboratory has been in continuous operation.

The need for the laboratory arose principally from the extraordinarily severe problems that faced the District engineers in connection with the pumping plants for the Colorado River aqueduct. This aqueduct will have a capacity of 1600 cfs, and in bringing this water the 300 miles from the Colorado River to Los Angeles and the other Southern California municipalities which have united to form the Metropolitan Water District, it is necessary to lift it a total of nearly 1700 ft. To do this will require about 350,000 hp, which classes it as the largest pumping project in existence. The location finally selected divides this lift between five pumping stations, working against average heads of from 146 ft for the lowest to 444 ft for the highest. Very little precedent was available for plants of such size, to assist the engineers of the District in answering questions concerning maximum permissible head per stage, single- or double-suction pumps, optimum speeds, attainable efficiencies, and desirable operating characteristics. It was felt that a properly equipped laboratory would be of great assistance in studying such problems, and would amply justify the expense required, both by savings expected and by the insurance of obtaining the most satisfactory type of equipment.

The responsibility of supervising the design, construction, and operation of the laboratory was placed in the hands of a group consisting of Professors Th. von Kármán, R. L. Daugherty, and R. T. Knapp for the Institute, and J. M. Gaylord, chief electrical engineer, and R. M. Peabody, senior mechanical engineer, for the district. In the two years since the construction was completed, the laboratory has been engaged in working on the following problems:

1 A comprehensive study of a group of pumps of varying specific speeds and other operating characteristics for the purpose of selecting the proper specifications for the pumps in the different stations.

2 Precision acceptance tests of both bidders' and contractors' model pumps.

3 A study of the transient flow characteristics of the contractors' model pumps in order to ascertain their behavior during emergency conditions such as power failure and shaft breakage.

4 Special internal surveys of the hydraulic conditions existing within the case, as they affect either structural design or operating efficiency of the unit.

5 Study of control-valve characteristics throughout the operating range from closed to full open.

6 Metering investigations.

The purpose of this paper, however, is not to discuss any of the details of the work of the laboratory, but is rather to give a description of the laboratory itself, its equipment and instrumentation, and thus furnish a foundation for subsequent reports dealing with the various investigations that have been undertaken.

General Description of Laboratory Circuits

(*A*) *Main Circuit.* The equipment in the laboratory is arranged in a series of closed hydraulic circuits, to increase its convenience and usefulness. The main circuit is shown in Fig. 1 and consists essentially of the low-pressure regulating tank, the machine under test connected to the dynamometer, the venturi meters, and the high-pressure service pumps. This circuit can be utilized with flow in either direction through the test machine, for by means of interconnections the suction and discharge connections of the service pumps may be reversed. Also, parts of the circuit not needed can be by-passed. An example of this is shown in Fig. 2, which is the circuit most used for the normal pump tests.

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(*B*) *Pressure-Regulating Circuit.* Superimposed on the main circuit are two auxiliary closed circuits, the arrangements of which can be seen in Fig. 3. The pressure-regulating circuit has a flow of about 450 gpm. Its function is to regulate the pressure in the low-pressure tank to any desired value between about 120 ft above atmospheric and 20 ft below. Since the only connection between the main circuit and the atmosphere is through this auxiliary circuit, it serves to stabilize the pressure level of the main circuit at the point desired. As the rate of flow through this regulating circuit is constant, the pressure in the tank is controlled by varying the resistance offered by the by-pass valve shown in Fig. 3. This valve is actuated from the operator's table. If the pressure desired is below atmosphere, it is necessary to run the by-pass pump to eject the regulating flow from the tank. In this case the vacuum pump located above the tank is also run to remove any dissolved air that comes out of solution in this region of low pressure. Since this pump operates through a barometric loop, no water leaves the system by way of this path.

(*C*) *Cooling Circuit.* The second auxiliary circuit shown in Fig. 3 is the cooling circuit. Fundamentally, all of the energy supplied to the machines in the main circuit is dissipated in heating the water. In some conditions of operation the dynamometer may be contributing 450 hp, the two high-pressure ser-

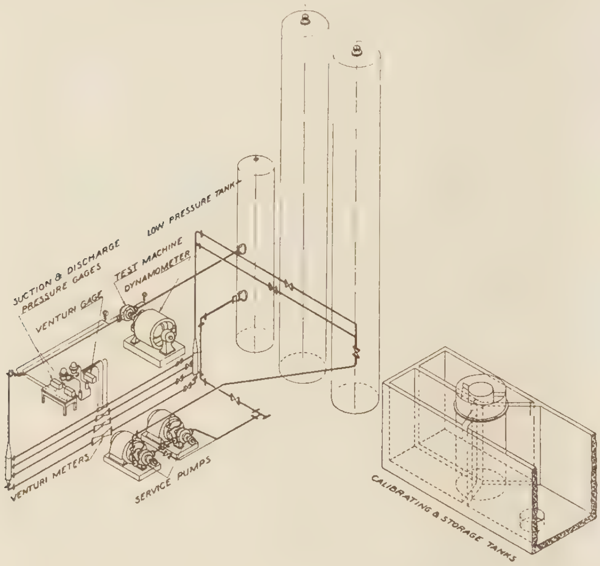

Fig. 1 The Main Circuit of the Laboratory

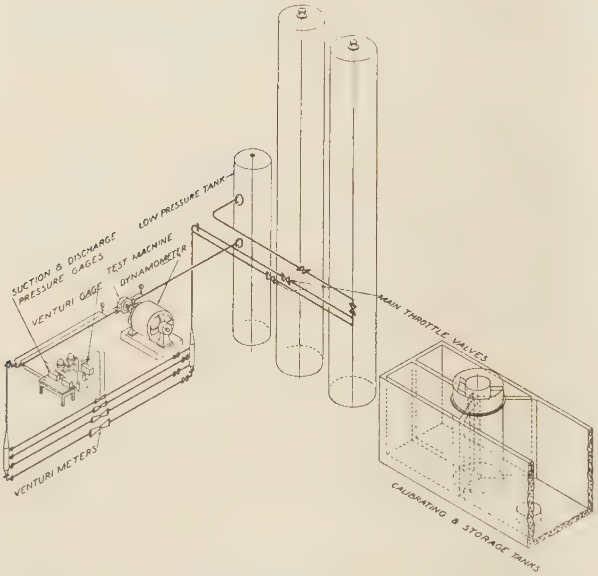

Fig. 2 Circuit Used for Normal Pump Tests

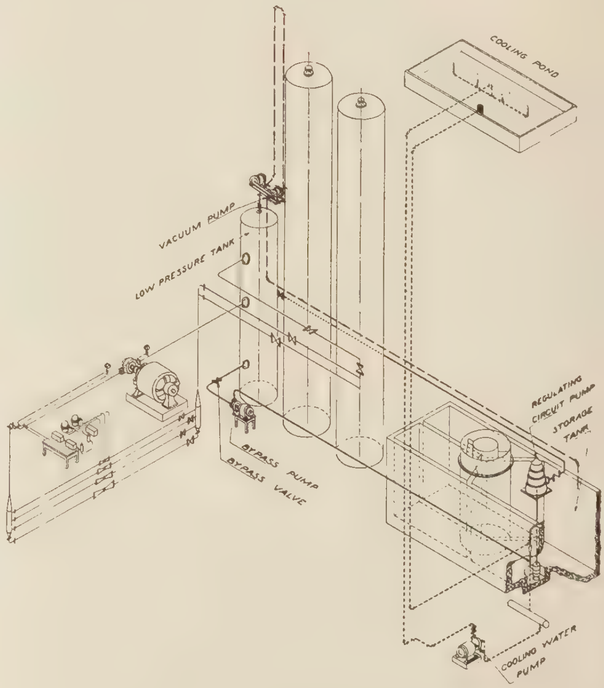

Fig. 3 Arrangement of Auxiliary Closed Circuits Superimposed on the Main Circuit

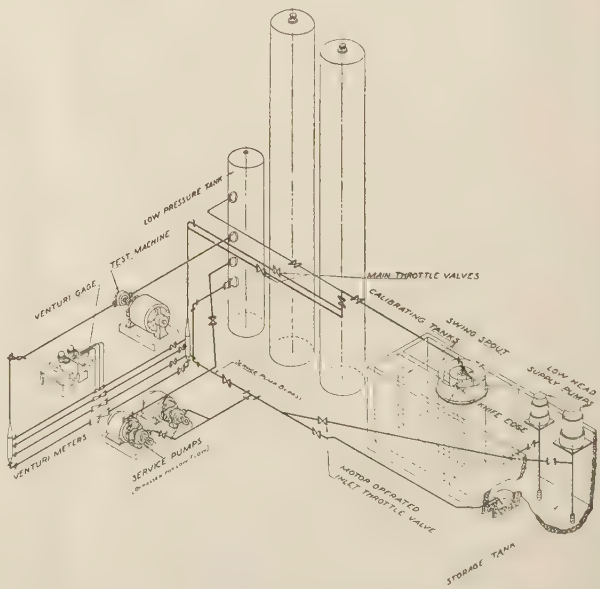

Fig. 4 Main Circuit Rearranged to Include Calibrating and Storage Tanks

vice pumps 250 hp each, and other sources say another 100 hp. The main circuit contains something less than 400 cu ft of water. Therefore, if nothing were done to control it, the temperature would rise about 2 deg per min. Since such a state would be intolerable, a portion of the flow coming out of the low-pressure tank through the by-pass line is diverted to a pump which sends it to spray nozzles in a pond on the roof. In this way enough heat is dissipated to keep the temperature under control for all conditions of operation.

(*D*) *Calibrating Circuit.* For the purpose of calibrating the venturi meters, or for making a direct volumetric determination of given points of operation of the machine under test, it is desirable to rearrange the main circuit as shown in Fig. 4 to include the calibrating and storage tanks. It should be noted that the venturi meters employed are of symmetrical construction so that they can measure flow in either direction. Although Fig. 4 shows the circuit arranged to calibrate one direction of flow, it is possible to reverse the flow through the meters without

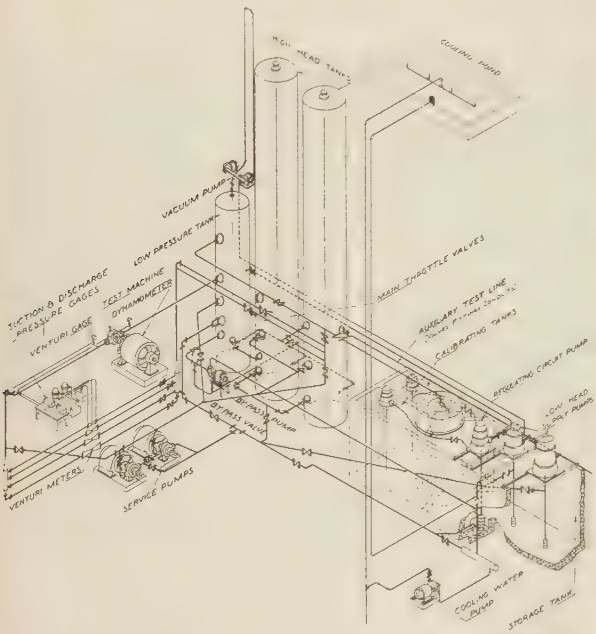

FIG. 5 COMPLETE PIPING DIAGRAM OF THE LABORATORY

changing the arrangement for calibration. Thus, the meters may be calibrated in place under all possible operating conditions.

(*E*) *Special Circuits.* Although the arrangements previously described are the ones used for the majority of the work, other circuit combinations are easily obtained. Fig. 5 shows the complete piping diagram for the laboratory, and may serve to demonstrate the versatility of the system.

GENERAL INSTALLATION

The main units of the laboratory equipment are housed in a room about 20 ft wide, 48 ft long, and 50 ft high. A working floor was constructed about 12 ft above the original level. The general appearance of this part of the laboratory is shown in Figs. 6 and 7. The basement formed below this floor houses the high-pressure service pumps, the by-pass valve and pumps, the venturi meters and other auxiliary apparatus. The calibrating tanks, storage tank, and supply pumps are in a long room that opens off from the northeast corner of the working floor. Thus, it will be seen that the entire installation is very compact and convenient to operate.

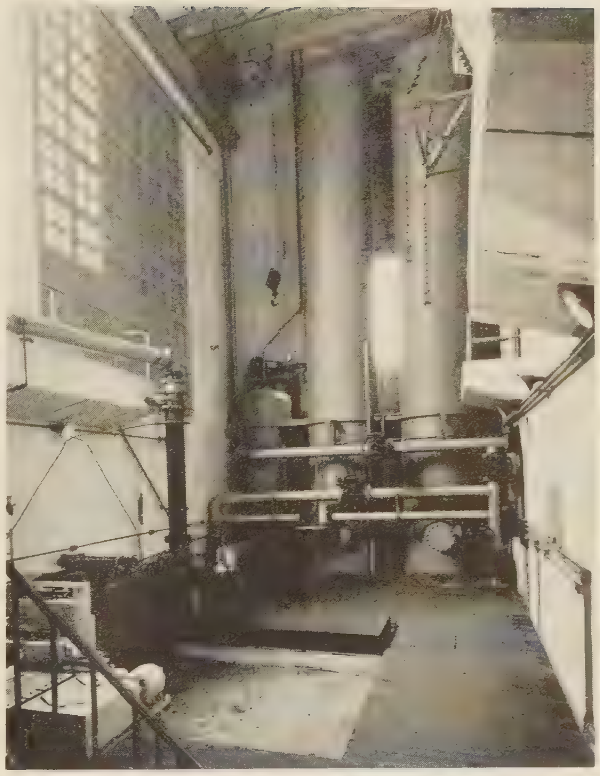

FIG. 6 WORKING FLOOR LOOKING NORTH, SHOWING PRESSURE TANKS IN THE BACKGROUND

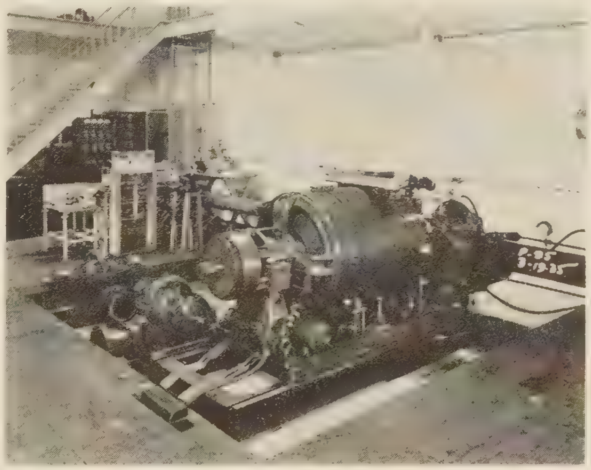

FIG. 7 WORKING FLOOR LOOKING SOUTH, SHOWING DYNAMOMETER IN THE FOREGROUND

LABORATORY EQUIPMENT

(*A*) *Dynamometer.* The dynamometer was furnished by the General Electric Company under special specifications prepared by the laboratory. It has a rated output of 275 hp as a motor, with ample overload capacity in reserve. In fact, it has delivered as much as 500 hp for short periods during the testing program. As a generator, its rating is correspondingly slightly higher. It can operate at speeds up to 5000 rpm, and has been tested to a runaway speed of 5500 rpm. It is, of course, a direct-current

machine and has a 250-v. normal operating rating. The separately excited fields are wound for 125 v. It may be operated equally satisfactorily in either direction of rotation.

The armature is provided with special high-speed ball bearings to eliminate the slight shift in radial position of the armature with respect to the frame which would have been present if sleeve bearings had been used. This shift would have been objectionable, since the variable unbalance accompanying it would have been considerably greater than the limits of accuracy of the torque measurements desired.

To provide the means for measuring the dynamometer torque, the frame is mounted on spherical roller bearings placed over

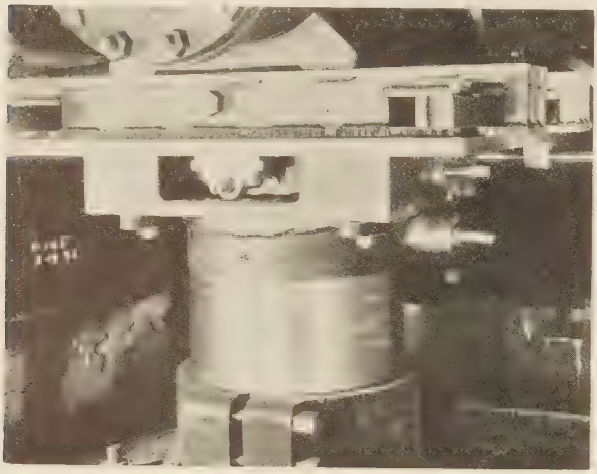

Fig. 8 Adjustable Base for Mounting Test Units

the shaft bearing housing. The outer races of these bearings in turn are mounted in sleeve bearings and are rotated slowly (about 7 rpm) in opposite directions by individual 0.5-hp motors. Thus, the static friction of these bearings is eliminated and the running friction is neutralized by balancing them against each other.

In addition, the possibility of these bearings "Brinelling" is eliminated, thus insuring the maintenance of the original sensitivity of the mounting to changes in torque. The frame is prevented from rotating by two stops fastened to the base. These are so adjusted that the dynamometer is free to rotate a total of only a few thousandths of an inch. No clamp is provided for use when starting or stopping, as it is unnecessary when the frame rotation is so limited.

A 5-kw, two-pole alternator is mounted on a shaft extension at one end of the dynamometer. The frame of this machine is fastened to the main-dynamometer frame; therefore, it has no effect on the torque reading of the dynamometer under any circumstances. The power from this alternator can be used to drive synchronous motors in any part of the laboratory which will therefore run either with the identical speed of the dynamometer, or at some definite fraction of that speed, as determined by the number of poles on the motor. One such motor is used to drive the tachometer and a contactor for giving a signal every 5, 10, or 50 revolutions of the dynamometer shaft.

The bottom and sides of the dynamometer base are carefully machined to fit the ways of two sub-bases. These are mounted on a massive concrete structure which is independent of the rest of the laboratory, this being done to reduce vibration to a minimum. One sub-base is mounted for use with machines having an axial pipe connection, such as single-suction pumps or axial-discharge turbines. The other sub-base is mounted at right angles to this position, for use with machines having both pipe connections at right angles to the shaft, such as double-suction pumps. The ways of the sub-bases provide for considerable axial adjustment in the position of the dynamometer, to meet the variations in dimensions of the different machines to be tested. The dynamometer is moved from one sub-base to the other by means of a traveling crane. The special bridle which has been constructed to facilitate this change may be seen on the right-hand wall in Fig. 6. The general appearance of the dynamometer is shown in the foreground in Fig. 7, while the two sub-bases may be seen in Fig. 6.

Since both the speed and the power involved are rather high, it has been thought advisable to provide a convenient emergency stop. Therefore, a system of overhead wires connected to a master relay switch is strung around the laboratory, so that in case any trouble develops the dynamometer can be shut down from any point on the operating floor.

The power supply for the operation of the dynamometer comes from a 700-kw motor-generator set which is part of the wind-tunnel equipment housed in the same building. Since it is not

Fig. 9 General View of Laboratory Basement Showing High-Head Service Pump

properly a part of this laboratory, no further description of it will be given here.

(*B*) *Adjustable Testing Base.* Installed on the same concrete platform with the dynamometer is an adjustable testing base on which the different machines under test are mounted. The principal adjustment provided is in the vertical direction, since there is considerable variation in the distance between the shaft and the base plate on the different machines submitted for test. Convenient adjustments of small range are also provided in the two horizontal directions and also in rotation. These latter movements greatly facilitate the precise lining-up of the machine to be tested with the dynamometer. A view of this base and the adjusting screws is seen in Fig. 8. The time required to change machines is reduced considerably by the use of this base, even though exact alignments must always be secured, due to the high speeds of rotation and powers involved.

(*C*) *Low- and High-Head Tanks.* The low-head tank and the two high-head tanks shown in Figs. 1 to 5, inclusive, are all of similar construction. They comply with the A.S.M.E. Unfired Pressure Vessel Code, Class I specifications, and are electrically welded, stress relieved, and all seams were X-rayed. In addition they were tested hydrostatically to 500 lb per sq in., although they are rated at 300 lb per sq in. working pressure. Their general appearance is shown in Fig. 6, although it must be remembered that they extend down to the floor below, making them about 12 ft longer than they appear in the illustration. The low-head tank has a volumetric capacity of about 350 cu

ft, while the high-head tanks each have a capacity of about 1000 cu ft.

(*D*) *Calibrating and Storage Tanks.* Two open calibrating tanks and a storage tank are provided. They are all 10 ft in depth and are built into a 10 × 10-ft concrete channel which was in the original building. In order to eliminate leakage, these tanks are lined with $^1/_8$-in. steel, welded in place and grouted to the original walls. The sides are vertical. The two calibrating tanks have capacities of 300 and 1000 cu ft, respectively, while the storage tank has a capacity of about 5000 cu ft. They are all provided with broad crested overflow wiers to prevent danger of damage to the rest of the laboratory and adjoining electrical equipment.

(*E*) *High-Head Service Pumps.* The two service pumps shown in the main circuit are located in the basement. They furnish the high-head water supply needed for reverse-flow tests, turbine tests, etc. They are single-stage, double-suction pumps and have normal ratings of 360 ft head and 2400 gpm. They are driven by 200-hp induction motors at a speed of 2900 rpm. They are installed on the foundation of the isolated dynamometer

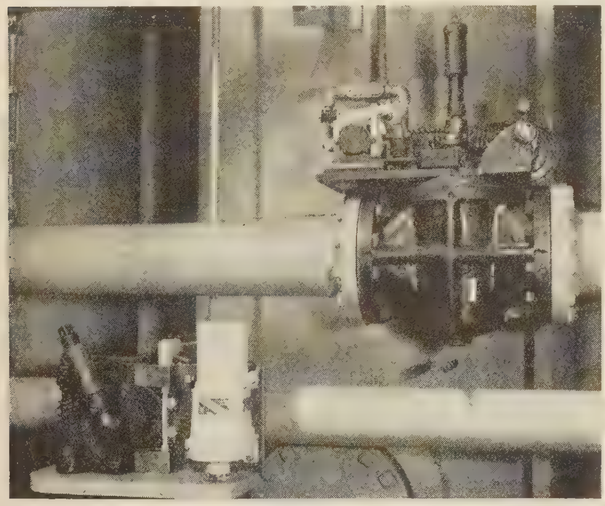

Fig. 10 Motor-Operated Throttle Valves Remotely Controlled From the Operator's Table

structure, as shown in Fig. 9. It is evident from the piping diagram shown in Fig. 5, that they can be operated either in series or in parallel, and that there are a number of alternate paths both for the suction and the discharge.

(*F*) *Low-Head Supply Pumps.* Two short-column deep-well pumps are submerged in the storage tank. Both develop heads of 50 ft and are arranged to operate either singly or in parallel. The smaller one has a capacity of 4 cfs while the larger has a capacity of 8 cfs. They are used to supply the intake head when a test pump is discharging to the calibrating tanks, to feed the service pumps when they are operating in parallel, or for other purposes where a relatively large quantity of water at low head is needed.

(*G*) *Auxiliary Pumps.* The supply and by-pass pumps of the regulating circuit are matched as to capacity, since they operate in series. They have ratings of 120 and 30 ft head, respectively, both at 450 gpm. The supply pump is of the four-stage short-column deep-well type and is submerged in the storage tank. The by-pass pump is of the simple single-suction close-built type with integral driving motor. Of its 30 ft total head, 20 ft or more at times may be suction-lift. Both these and all the other auxiliary pumps are induction-motor driven.

The calibrating-tank pumps are used to empty these tanks after a calibrating run has been made. It was felt undesirable to have any openings in these tanks because of possible leaks. Therefore, these pumps are again simply short-column deep-well types which are suspended from above and discharge over the tops of the side walls of the calibrating tanks. Incidentally, the one in the larger tank has an adjustable-pitch propeller which gives the possibility of carrying on some interesting work with it in the future. With its present setting, it has a capacity of about 5000 gpm against a 12-ft head.

The cooling-water pump is of the simple single-stage type and has a capacity of 150 gpm against a head of 120 ft. The vacuum pump is a Nash Hytor, and can be seen mounted above the low-pressure tank in Fig. 6.

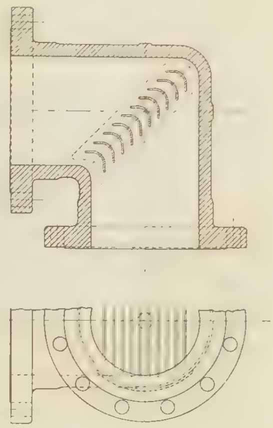

Fig. 11 Section Through the Cast-Iron Vane Elbow

(*H*) *Valves.* For convenience, it was decided to operate several of the valves by motor, and to control them remotely from the operator's table. Grease-lubricated plug cocks were utilized for this purpose, with special diamond ports for the main throttle valve and the by-pass valve. The normal positive stops were removed, limit switches installed in their places, and a 0.25-hp motor with integral gear reducer and electric brake was connected by chain to the hand-wheel shaft of the valve mechanism. Fig. 10 shows the two throttle valves in position.

Grease-lubricated and sealed plug valves were also used on the bank of venturi meters, because they were apparently the most leak-proof under operating conditions and it was very necessary that there should be no leakage flow through the meter lines not being used. However, these valves are manually operated since they are used only when changing meters.

Gate valves were used in the remainder of the piping system. Of these, some were in key positions where leakage through a closed valve would affect the accuracy of the measurements, while others were so placed that leakage was of small moment. For the key positions double-disk valves were employed, with special bleeder connections between the disks. No leakage from an open bleeder on a closed valve is positive assurance that there is no leakage through the valve.

(*I*) *Vane Elbows.* In several locations in the laboratory piping system it was desirable to have as little disturbance as possible downstream from an elbow. For this purpose a series of vane elbows were designed by the laboratory, patterned after wind-tunnel practice. Two types were constructed, one a cast-iron casing with steel vanes placed in the core, and the other of all-welded construction. Fig. 11 is a section through a cast elbow, showing the vane spacing which is typical for all sizes. The cast type was made in the 10-in. size only, while the welded construction was employed for 8-, 12-, and 16-in. sizes. Figs. 12 and 13 show the appearance of the finished elbows. It should be noted that it is possible to secure a most compact construction with the vane-type elbow.

Laboratory Instrumentation

Before the instruments were designed, a thorough study of the needs of the laboratory was made. This resulted in the following general specifications for the entire group of instruments:

1 Precision and sensitivity. The work contemplated called for an accuracy of an individual reading of 0.1 per cent.

2 Elimination of personal equation. The necessity of using several different operators made it desirable to have the readings as impersonal as possible.

3 Primary standard type of instruments. Part of the work contemplated for the laboratory was the testing of model pumps, supplied by different bidders or contractors. Due to the variety of interests involved, it was felt that instruments whose accuracy depended on fundamental measurements of length, weight, and time were to be preferred, where possible, to those of the secondary standard type whose accuracy depended entirely on calibration.

Fig. 12 The 10-In. Cast-Iron Vane Elbow

4 Speed of measurement. It was known in the beginning that there was a great deal of work to be done by the laboratory in the limited time available, therefore an instrument ensemble which could speed up the observations without sacrificing accuracy would be very advantageous.

5 Flexibility. Although certain tasks were definite at the start, the entire program could not be outlined ahead of time, because the project was fundamentally a research undertaking. For this reason it was decided to strive for as much versatility as compatible with the known objectives of the laboratory.

Quantities to Be Measured or Controlled. The fundamental quantities for which instruments were to be designed to measure or control were speed, torque, inlet and discharge pressures, and rate of flow. During the course of the work other instruments were developed to measure special properties, but they do not properly belong to the primary equipment of the laboratory and will not be described here.

Speed. In working with high-speed hydraulic machinery, probably the measurement and control of speed causes the most difficulty. Not only is it hard to measure in itself, but the slight variations present with most equipment are reflected in the torque, head, and flow readings as well. Therefore, it was decided to endeavor to construct a speed-control system to hold the test machine at the precise speed at which the tests were desired, independent of fluctuations of load or other disturbances. The basic principle finally adopted is a comparison between a known standard speed and the speed of the machine under test, with any existing difference, no matter how small, acting to correct the speed of the test machine.

Standard Reference Speed. The primary accuracy of such a system depends first upon the accuracy of the reference speed. The first thought for a source of such a reference is naturally a synchronous motor driven from the local power supply. The accuracy of this source over long periods of time is unquestioned, since it is used to drive clocks which are never out more than a total of a few seconds in 24 hr, which is a precision much greater than the 0.1 per cent set for the laboratory instruments. However, a more careful investigation showed that the situation was not so favorable. Short-time variations of 0.75 per cent were found to be relatively common, lasting for periods of from a few seconds to several minutes. Since this condition was not tolerable, it was decided to construct a 1-kw standard-frequency system, the power of which could be used to drive the standard reference motor, a chronograph, and other auxiliaries which might demand an absolutely constant, known speed.

For the basis of this system a 40-kilocycle quartz crystal was chosen, provided with a thermostatic case. This frequency is picked up in the usual manner with a loosely coupled vacuum-tube circuit and then stepped down to 800 cycles through a pair of multivibrators acting in series. This is then amplified until a few watts of power are available and a special 800-cycle synchronous motor is driven by it. This in turn operates a 50-cycle commutator which is used to control a thyratron inverter circuit. The output from this is 1 kw of 50-cycle single-phase power at 110 v.

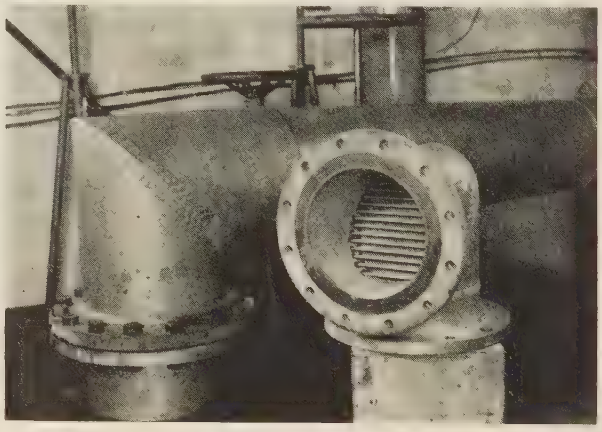

Fig. 13 The 12-In. Welded Vane Elbow

There are two simple checks of the accuracy of this system. In the first, the frequency of the quartz crystal is compared with the carrier frequency of one of the several local radio stations. This carrier frequency is claimed to be accurate and constant to one part in three hundred thousand. A convenient little circuit employing one of the small cathode-ray tubes, now so popular for silent tuning of radios, serves to visually indicate the beats between the two systems. By this means it has been determined that the laboratory standard-frequency system has a minimum accuracy of one part in one hundred thousand. The second check is a rough one and is used primarily to insure that the multivibrators are operating on the proper steps. It consists in simply comparing the 50-cycle output frequency from the inverter

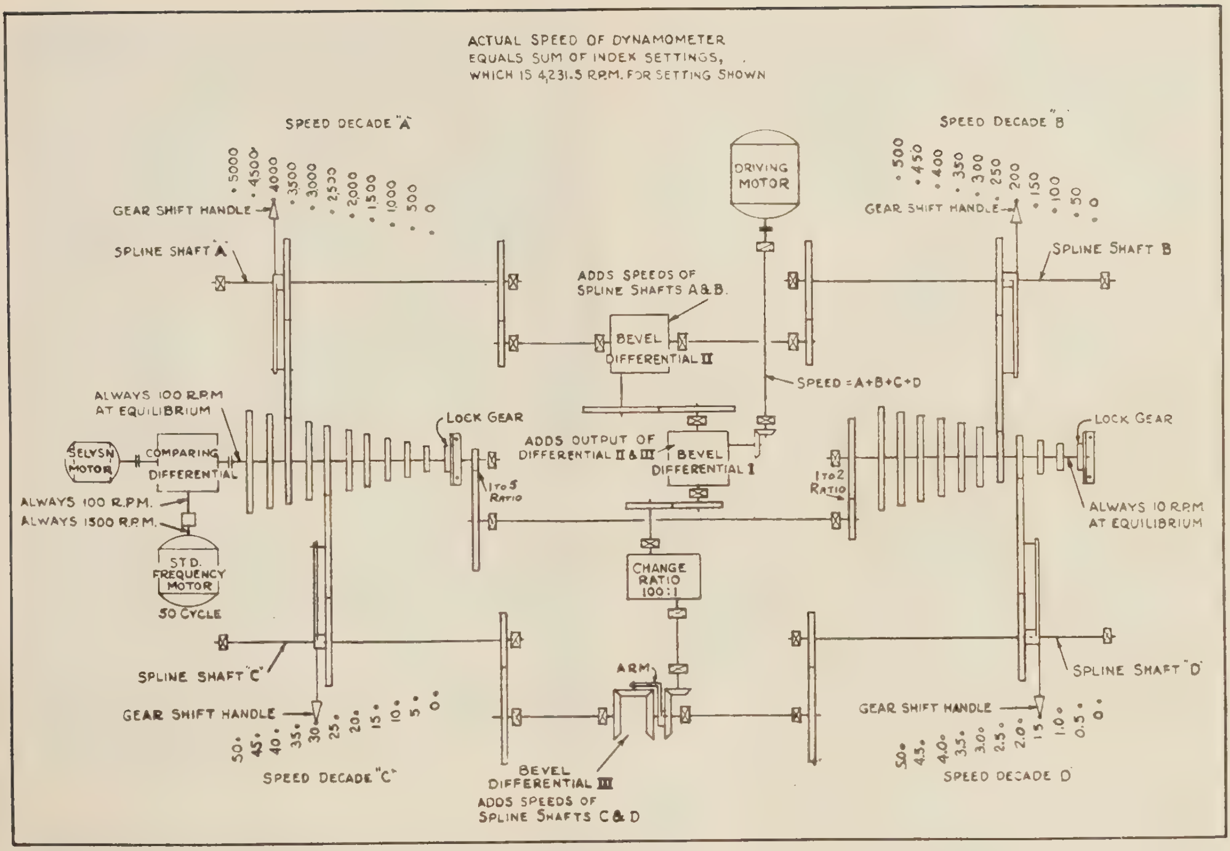

Fig. 15 Diagram of Gearbox Internal Arrangement

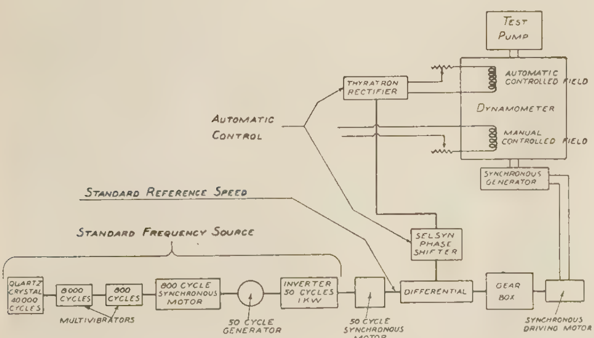

Fig. 14 The Dynamometer Speed-Control System

with the local 50-cycle power line. Incidentally, this gives a constant check on the deviations of the local power and has shown that the laboratory was amply justified in constructing its own standard-frequency system.

Dynamometer Speed Control. The speed-control system, shown in Fig. 14, is built around two synchronous motors, one operated by the standard-frequency system and the other by the alternator on the dynamometer shaft. These drive two shafts of a small bevel-gear differential. The third shaft therefore turns at a speed proportional to the difference of the other two. This shaft actuates a phase shifter (in this case a selsyn motor driven through a friction clutch and limited in motion by stops) which controls the output of a battery of thyratron rectifiers. These furnish the excitation field for the shunt-wound dynamometer, and thus control its speed. It will be seen that any difference in speed between the dynamometer and the speed standard acts immediately to correct itself. The only position of equilibrium is absolute synchronism of the two systems.

One detail of the field control is of interest. The field windings on each pole are split into two equal coils, thus giving two equal field circuits. Only one of these receives its excitation from the controlled thyratron rectifiers. The other goes to the normal direct-current laboratory exciter bus. By means of field rheostats, the relative strengths of the fixed and variable portions of the field windings are adjusted easily to eliminate any tendency for hunting to occur.

Interposed between the control differential and the dynamometer-driven synchronous motor is a multirange gearbox. With this it is possible to adjust the ratio between the standard reference speed and that of the dynamometer so that the latter may be operated at any speed between 1000 and 5000 rpm in one-half revolution steps. This is accomplished with four gear-shift levers by use of differentials which permit the addition of ratios rather than simple multiplication.

Fig. 15 is a line diagram of the internal arrangement of the gearbox. With this construction each shift handle controls a decade, the steps of which are 500, 50, 5, and 0.5 rpm, respectively, which makes the setting of any chosen speed a very simple matter.

The control shaft operating the phase shifter is provided with a pointer. When this pointer is stationary, irrespective of location, it indicates that the dynamometer speed corresponds exactly to the gearbox setting. This is all the speed measurement that is necessary. Chronographic checks have demon-

strated that this system is very reliable, and that the maximum instantaneous variation from the set speed under steady load conditions is about 1 rpm.

Fig. 16 is a view looking down upon the gearbox with the cover removed to show the construction, and Fig. 17 shows the appearance of the assembly installed at the operator's table.

Torque Measurement. The method of mounting the dynamometer for weighing the torque reaction has already been described. The sensitivity of this mounting proved to be 0.01 ft-lb.

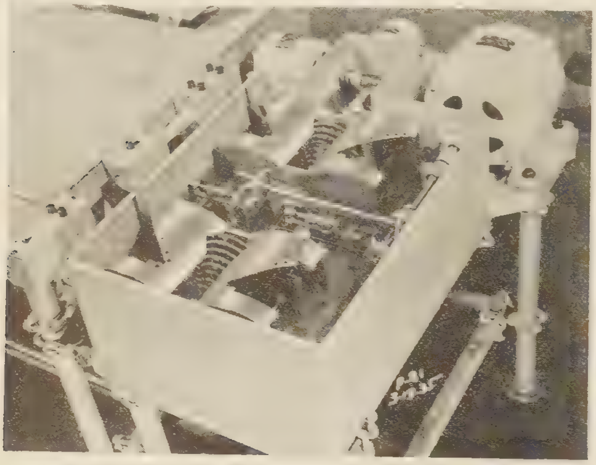

Fig. 16 Gearbox With Cover Removed

Fig. 17 The Gearbox Installation

In order to measure the torque accurately, two systems were installed, the lower and the upper weighing mechanisms. The lower weighing mechanism is manually operated and measures units of 100 ft-lb only, while the upper weighing mechanism is automatically balanced and weighs to the nearest 0.01 ft-lb, with a maximum of about 125 ft-lb. Together they have a capacity of 1200 ft-lb, which is sufficient for any condition of operation of the dynamometer.

Lower Weighing Mechanism. The lower weighing mechanism operates hydraulically. It consists of two ground and lapped pistons and cylinders mounted on the dynamometer base and working against knife-edges on the dynamometer, a smaller cylinder and piston of similar construction on the operator's table for applying the loads, a transmission line between them, and a pump for

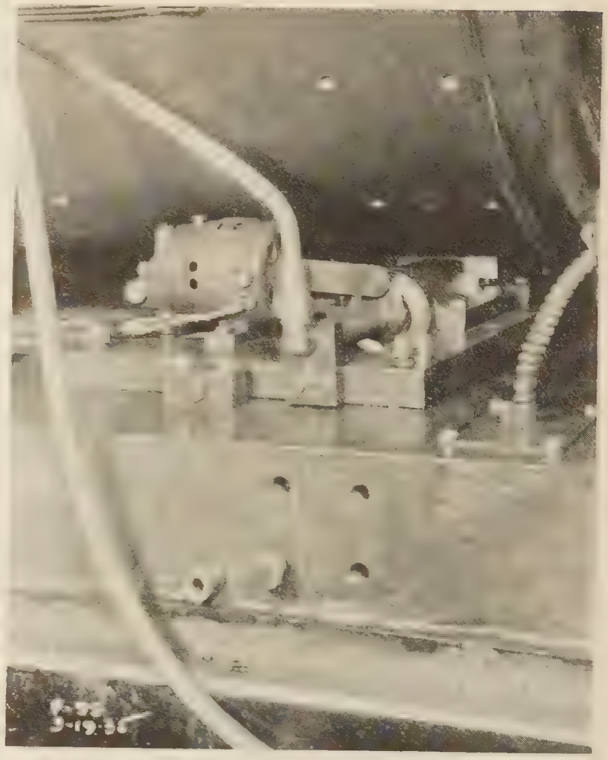

Fig. 18 Operating Portion of the Lower Weighing Mechanism Mounted on the Dynamometer

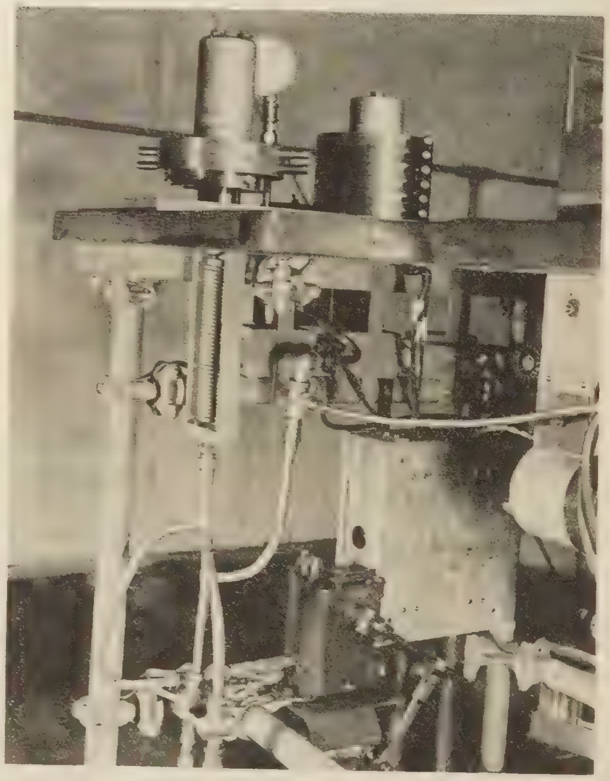

Fig. 19 Control Portion of the Lower Weighing Mechanism, Mounted on the Operator's Table

periodically supplying small quantities of oil to the system to take care of the leakage. The two sets of pistons and cylinders are required on the dynamometer to take care of both directions of rotation, but only one is in use at a time. The cylinders are all oscillated continuously by small motors, which eliminates wall friction and increases the sensitivity. The two dynamometer units are so accurately paired that there is no detectable difference in their readings. Although the pistons are about 1.25 in. in diameter, and the working clearances are the order of 0.0001 in., they can be interchanged in the cylinders.

Fig. 18 shows part of the dynamometer installation and Fig. 19 shows the parts that are on the operator's table. Note that the leakage supply pump is a stock high-pressure positive lubricator equipped with a motor drive which is operated from a push button conveniently located on the table. The stock of 100-ft-lb weights are to be seen on the table to the right of the piston and cylinder assembly. The oscillating motor is mounted close under the table top and is visible at the right of the flexible helix leading up to the cylinder.

Upper Weighing Mechanism. The upper weighing mechanism is very simple in operation and is mounted directly on top of the dynamometer. It consists of a calibrated weight which moves horizontally, normal to the axis of the dynamometer. It is driven by a precision screw which is carried in preloaded bearings to eliminate any back-lash. Therefore the revolutions of the screw measure the torque, and all that is necessary is to attach a revolution counter through a proper gear train to give the reading directly in foot-pounds. This is relayed to the operator's table with a pair of selsyns to increase the speed of testing. Fig. 20 shows the installation. The screw is driven by a reversing motor, which is controlled by sets of contacts between the

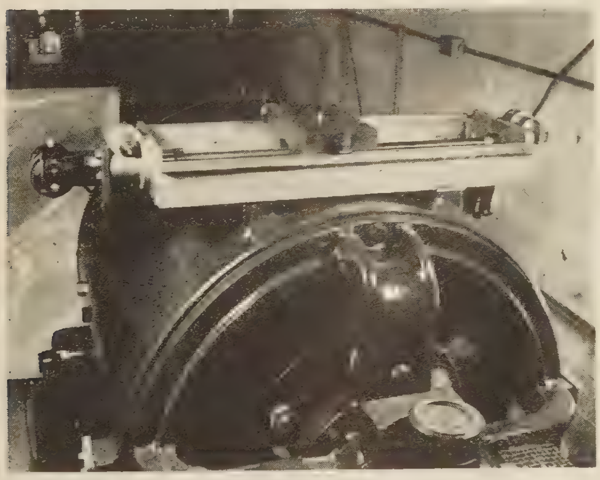

Fig. 20 The Upper Weighing Mechanism

dynamometer frame and the base. The 0.003- to 0.004-in. freedom of oscillation permitted the dynamometer by the frame stops is sufficient to operate the contacts. Sparking and welding are prevented by use of vacuum-tube relay circuits, which also permit of an adjustable time delay which is used to prevent hunting. The same contact and relay system also operates solenoids of adjustable strength which apply small countertorques to the dynamometer frame. This, in combination with the time delay, has been successful in eliminating all hunting of the automatic balancing mechanism.

The calibration of both upper and lower systems is accomplished by hanging known weights directly on the dynamometer

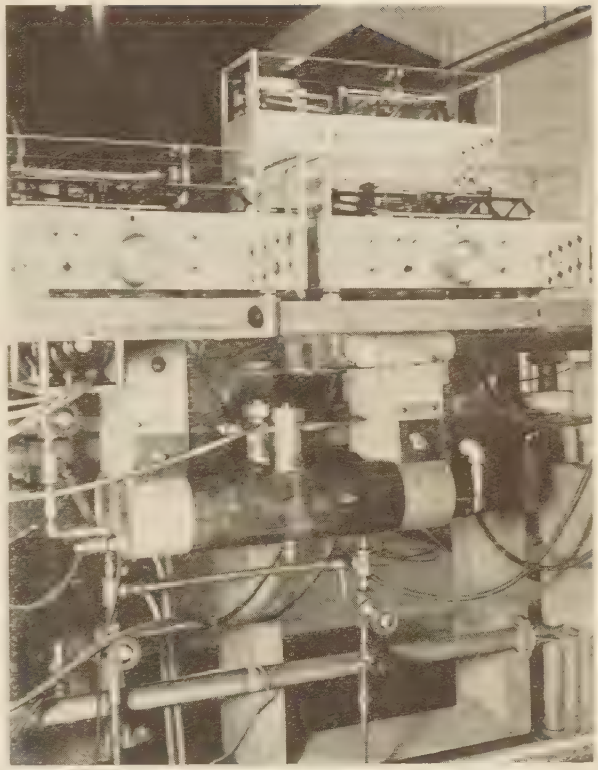

Fig. 21 Oil-Operated Pressure Gages With Oil Reservoirs Mounted Beneath

on knife-edges supplied for that purpose. The accuracy of the combined system considerably exceeds the 0.1 per cent limit originally set as the goal.

Head or Pressure Measurement. In working with hydraulic machinery, it is of course necessary to measure heads in terms of feet of the actual fluid used. However, since this varies considerably with temperature, the gages designed for the laboratory measure in pressure units instead, i.e., in pounds per square inch. In principle they are pressure-weighing scales, since they weigh the force the pressure exerts on a piston of known size. They are constructed with a single beam mounted on special scale ball bearings. On one end of the beam there is a weight pan and on the other a dashpot and two sets of contacts for operating the motor-driven rider that runs along the top of the beam. The force from the pressure piston is applied to the beam through another set of pivot bearings. This piston is ground and lapped and operates in a cylinder of the same construction, which is rotated by a small induction motor. The pressure is brought to the cylinder through another lapped fit at the opposite end.

Ten weights, each equivalent to 50 lb per sq in., are provided for the weight pan. Each one is carried in an individual frame and can be lowered onto or removed from the weight pan by a small lever on the front of the gage case. A small Bourdon gage of the conventional type serves to indicate the proper number of weights to apply to bring it within the self-balancing range of the rider. An extra weight is provided which normally rests on the pan. When this is lifted, the gage measures pressures from a hypothetical level 50 lb per sq in. below atmospheric pressure. To prevent air from being drawn into the system when reading negative pressures, a small oil reservoir is provided around the piston where it emerges from the cylinder.

The motor drive for the rider is mounted on the case instead

of the beam. The rider itself is moved by a fine wire stretched along its length and located exactly in line with the pivot center of the beam. Thus, a force along the wire produces no torque on the beam and does not affect the balance. Clamped to the center of this wire is a split nut which runs on a precision screw, also mounted in preloaded bearings. This screw is driven by a small reversing induction motor running in oil under the case, which is controlled by the beam contacts through vacuum-tube relay circuits similar to those used on the upper weighing mechanism. The revolution counter driven by the screw gives the rider reading in hundredths of a pound per square inch. The inlet- and outlet-pressure gages are identical in construction and can be seen in the upper part of Fig. 21. Two balance lights and a cylinder-motor pilot light are seen on the front of the case. There are also switches for the manual operation of the rider and one which disconnects the contact relays from the beam motor and permits their use to control a motor-operated valve in case it is desirable to maintain a pressure at a fixed value.

To prevent deterioration of the lapped surfaces, the gages are operated with oil. Each one has a large oil reservoir directly

Fig. 22 Swing Spout and Distributing Chute on the Calibrating Tanks

beneath it as shown in Fig. 21. Any possibility of grit entering the cylinder is prevented by a fine-mesh strainer located in the turret on top of the reservoir. The reservoir is horizontal to permit a large volume change with a small level change in the oil, since the reading will be affected by the difference in density of the oil and water and the change in level. At the time of construction there was no precise knowledge of the rate of leakage of oil through the gage. Experience has now shown that the reservoirs have capacity for several years' operation per filling.

The primary calibration of the gages was simply the measurement of the piston and cylinder diameters and the lengths of the lever arms from the beam fulcrum to the piston pivot and to the weight-pan pivot, together with the weighing of the rider and the weights. This was also true for the primary calibration of the upper and lower torque-weighing mechanisms. The pressure gages were further checked by comparison with a Crosby deadweight gage tester, and an even more sensitive check was obtained by connecting them to the line of the lower weighing mechanism. The zero-reading balance is made by connecting them to an open riser filled with water to the level of the top of the gage cylinders.

Measurement of Rate of Flow, Primary Standard. The primary standard for measurement of rate of flow in the laboratory is the pair of volumetric calibrating or measuring tanks already described. They are used in conjunction with a swing spout and distributing chute so constructed that the flow suffers no disturbance when it is switched from one tank to the other. Fig. 4 shows the circuit employed.

Fig. 22 shows swing spout and distributing chute, with the

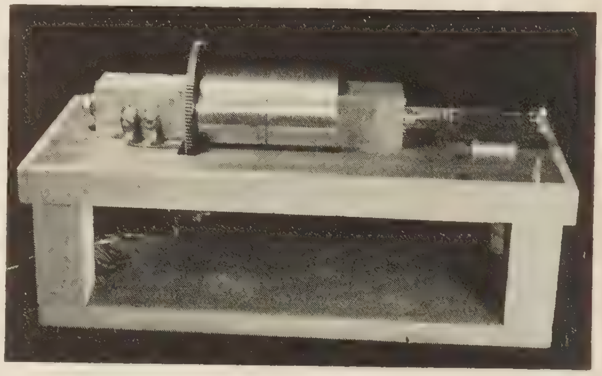

Fig. 23 Chronometer for Recording Time to Fill Calibrating Tanks

Fig. 24 Standardizing Pipettes and Measuring-Tank Point Gages

swing spout in position to discharge into the large calibrating tank. Note the use of the vane elbow at the entrance to the swing spout to insure even distribution and insensitivity to the position of the spout.

The common walls of the two calibrating tanks and the storage tank intersect at angles of 120 deg. The distributing chute is centered directly over their common intersection, and one third of it discharges into each tank. The swing spout is directly over the chute, and pivots about its center. It is driven by a constant-speed induction motor with integral gear reducer and electric brake. Limit switches on the rim of the chute permit it to travel only one third of a revolution before it is automatically stopped, i.e., the distance necessary to switch the flow from one tank to another. It is operated in either direction at will by push-buttons on the adjacent wall. The spout clears the knife-edged chute partitions by about 0.25 in., so a very clean cutoff is obtained. When the center of the spout is directly over the partition, a contact is made which actuates a special chronometer. Thus, the exact time of filling of a measuring tank is recorded. The time of passage across the partition is about 0.16 sec. Runs of less than 1 min are not taken. Therefore, inaccuracies due to dissimilarity of flow conditions in the spout when entering and leaving the tank must be of very small order.

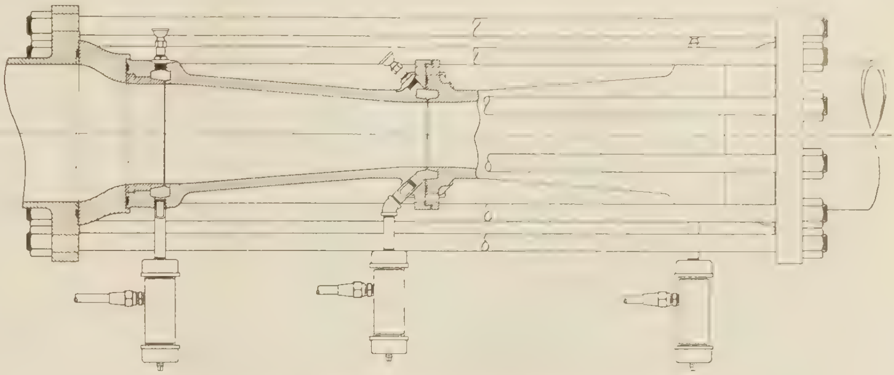

Fig. 26 Cross Section of the Venturi Meters

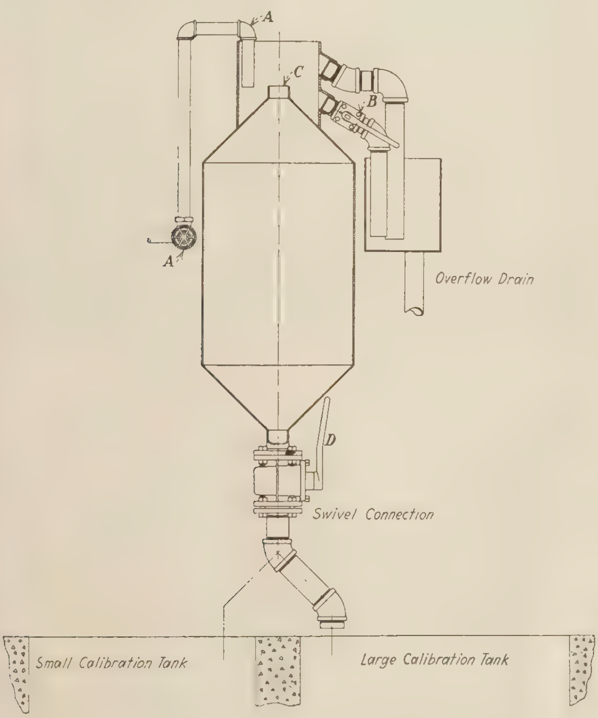

Fig. 25 Cross Section of the Standardizing Pipette

Chronometer. The special chronometer shown in Fig. 23 is used with these tanks. It is simply a revolution counter driven by a synchronous motor which is operated from the standard-frequency system. In use, the motor runs continuously but the counter is started and stopped by a pair of powerful magnetic clutches operated through surge circuits by the swing-spout contacts. The inertia of the counter mechanism is so low in comparison to the power of the clutches that the readings obtained are accurate to the nearest 0.001 sec.

Point Gages. The changes in level in the calibrating tanks are measured by point gages permanently mounted in each tank. These consist of a standard stainless-steel tape carrying a relatively heavy brass plummet which has a small platinum point protected by a bakelite ring. The tape is connected to the ungrounded side of the alternating-current lighting circuit through a neon glow lamp. As the point touches the water surface, the lamp lights, thus furnishing a sensitive method of locating the surface. The tape is read with a vernier, the smallest division of which is 0.001 ft. No difficulty is encountered in repeatedly duplicating readings to this precision. The installation is shown on the right in Fig. 24. The unused tape is wound on the reel seen on the side of the column. This is fastened with friction disks to the slow-motion shaft of a small commercial speed reducer. The knob seen in front is on the high-speed shaft and furnishes a micrometric slow motion for the tape.

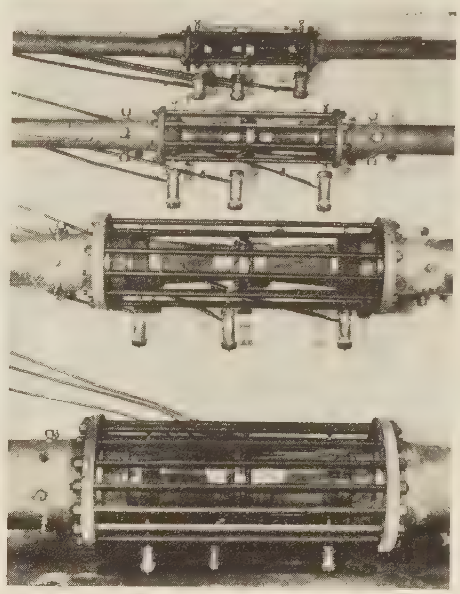

Fig. 27 The Installation of the Venturi Meters

Standardizing Pipettes. To standardize the measuring tanks a pair of pipettes were constructed. They are shown on the left

in Fig. 24. Fig. 25 is a sectional elevation of one of the pipettes and shows the method of operation. The valves D and B are first closed. The pipette is then filled by admitting water through A into the filling ring, where it overflows crest C and enters the pipette body. When it is full, a slight additional rise causes water to run out of the overflow pipe. At this signal the operator closes valve A, opens B and drains the ring, thus leaving the pipette exactly full. Valve D is now opened and the pipette discharged into the measuring tank. A standard draining time of 5 min is allowed, after which the process is repeated and the next calibrating point secured. Since the small pipette holds about 3 cu ft, and the large one 10 cu ft, while the tanks hold

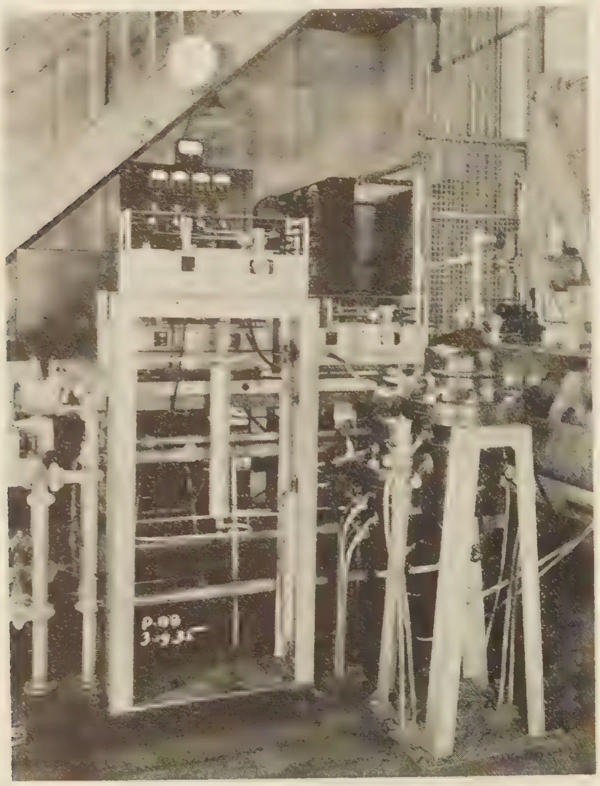

Fig. 28 Rear View of Venturi Manometer and Sediment Pots

300 and 1000 cu ft, respectively, it is possible to secure 100 points on each tank-calibrating curve. The pipettes themselves were standardized by weighing them both full and after a standard drain while they were suspended from one of the wind-tunnel balances. This balance was checked against standard weights and has a sensitivity of about a hundredth of a pound. A density determination of the water used was made at the temperature of the calibration.

Venturi Meters. Although always available for use, the measuring tanks require too much time to be convenient for normal testing. Therefore, a bank of four venturi tubes is provided as a secondary means of measuring rates of flow. These are graded in size so that each covers a range of only three to one, thus insuring a minimum of 3 in. of mercury differential pressure on the lowest rates permissible with each meter. Each meter is symmetrical and is provided with three piezometer rings, one on each end and one at the throat. Therefore, it is possible to measure flow in either direction by the proper selection of pressure connections. Fig. 26 shows a cross section of a typical tube and Fig. 27 shows the tubes installed. It will be seen in Fig. 26 that the piezometer opening is an annular slot. Its width is one tenth of its depth. This corresponds to accepted aerodynamic practice for accurate pressure measurements. Stuffing-box fittings are shown on the larger pipe just ahead of the venturi tubes in Fig. 27. These are for making velocity traverses with direction-finding tubes, which are being carried on in connection with a study of the tube coefficients.

Venturi Differential Manometer. The differential pressure from the venturi tubes varies from 3 to 30 in. of mercury. This is measured by the weighing-type differential mercury manometer shown in Fig. 28. Advantage was taken of the design developed for the pressure gage by using the mechanism in its entirety to weigh the low-pressure leg of the differential manometer. The piston and cylinder were omitted, the fulcrum point interchanged with the piston pivot point, and the manometer tube was suspended from the previous location of the fulcrum. The suspended

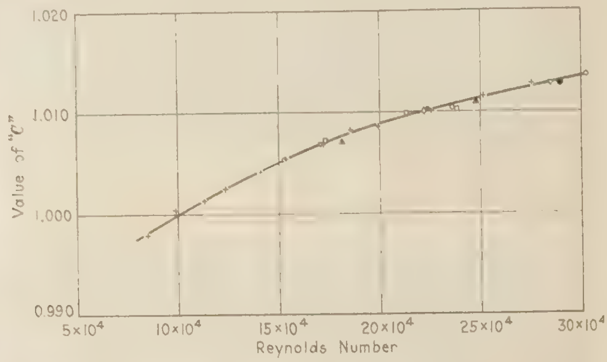

Fig. 29 Calibration Curve for the 12-In. Venturi Meter

manometer tube is connected to the other leg by a horizontal loop of thin-walled tubing about 3 ft long. On the beam, the contacts have a total clearance of about 0.003 in., and the manometer suspension point has about one tenth their lever arm, thus giving it a motion during weighing of about 0.0003 in. The force required to deflect the connecting tubes this minute amount has been found to be negligible. The manometer proper is constructed entirely of stainless steel. The sensitivity of the system is about 0.0006 in. of mercury.

Sediment Pots. Interposed between the venturi tube and the differential manometer are two sediment pots, one for each leg. They serve as headers as well, for all four throat connections come to one, and all eight end connections come to the other. Each line is provided with a ground cock, so that by proper selection any meter can be connected to the manometer to measure flow in either direction. These sediment pots are shown on the right-hand side of Fig. 28. It will be seen that they are in two parts, connected with a flange joint. The lower flange also serves to support a separating plate, on the bottom side of which are 12 short tubes, about 1.25 in. diameter, brazed to corresponding holes in the plate. Fastened over each tube and suspended from it is a very thin rubber bag. Upon assembly, the bags in the throat pot are collapsed with the lower half full of water, while those in the end-connection pot are nearly filled to capacity, although care is taken to insure that there is no distention of the rubber. The upper halves of the pots are then put in place and the manometer side of the system filled with distilled, deaerated water. Thus, the manometer operates in a closed system with no chance for dirt to enter it and collect in the mercury surfaces. However, since the combined volume of the rubber bags in each pot is considerably greater than the total volume of mercury displaced, no pressure difference can exist across the seal.

Venturi Calibration. Each venturi tube is calibrated by the measuring tanks in the manner described previously. Since it is possible for the water temperature in the laboratory to vary considerably under different testing conditions, the meter coefficients are plotted against Reynolds numbers instead of rate of flow. One of the typical calibration curves is shown in Fig. 29. The meter calibration is very constant, as shown by calibration runs taken at various times during the work. This is not surprising, since the meters are constructed entirely of bronze, and the approach lines are galvanized.

Operators' Table. From the foregoing description it is evident that the heart of the laboratory is the operator's table. Two views of this are seen in Figs. 30 and 31. The former shows the relative location of the different measuring instruments and controls. The latter shows the convenience of the operator's position with reference to such equipment as the main control board, the standard-frequency cabinet, and the thyratron installation.

It has been seen that the dynamometer, the pressure gages, and the venturi manometer are all self-balancing, and are operated

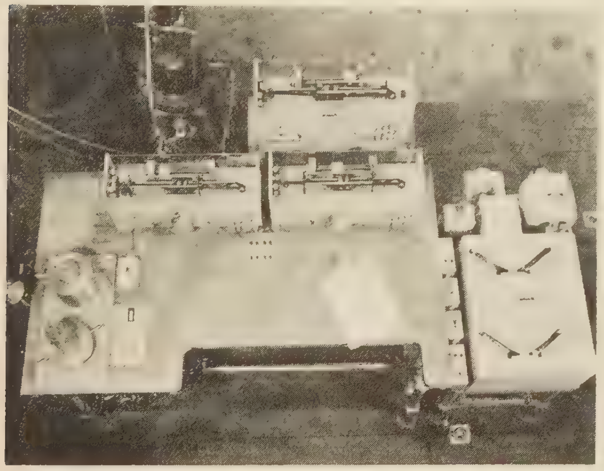

FIG. 30 TOP VIEW OF OPERATOR'S TABLE

by small motors. In order to secure simultaneous readings and to speed up the work, all of these balancing motors operate from a common circuit, the control switch of which is placed in a location convenient to the operator's hand. When making a run, he therefore simply watches the pilot lights from these instruments and when they all indicate a balance he opens this master switch and stops all of the balance motors. He then makes his adjustments for the next desired condition, and, while the new flow is coming to equilibrium, he records the previous measurements, as read on the various revolution counters. This accomplished, the switch is again closed, and in a few seconds all of the instruments are again indicating a balance. During all this time the speed-control system has been holding the dynamometer speed absolutely constant.

Overall Accuracy of Measurements. Table 1 gives a summary of the accuracy of measurements for the different principal readings. It shows that even when all of the available sensitivity is not employed each principal quantity is measured with an accuracy of better than 0.1 per cent, the goal originally set. The laboratory personnel feel, however, that for normal work a combined figure of accuracy of 0.1 per cent is about what is justified, because of minor instabilities in the flow itself which tend to lessen the precision. On the other hand, on several occasions the staff has tried to blame unexpected apparent irregularities of performance of individual test machines on the testing

TABLE 1 ACCURACY OF PRINCIPAL LABORATORY READINGS

		Sensitivity		Smallest normal reading	
Reading	Range	Maximum	Per cent of average reading	Reading	Per cent of average reading
Speed	1000 to 5000 rpm	1 rpm[a]	0.033	1 rpm[a]	0.033
Torque	0 to 1200 ft-lb	0.01 ft-lb	0.0025	0.1 ft-lb	0.025
Pressure	−15 to 550 lb per sq in.	0.01 lb per sq in.	0.007	0.1 lb per sq in.	0.067
Rate of flow	3 to 30 in. Hg	0.0006 in. Hg	0.002	0.006 in. Hg	0.02

[a] Fluctuation about mean value —not deviation from reading

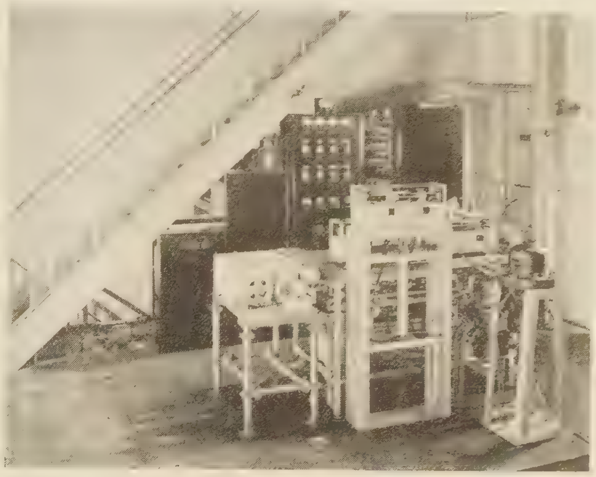

FIG. 31 REAR VIEW OF OPERATOR'S TABLE

equipment, but in each case rigorous investigation has absolved the instruments from suspicion of error and has proved that the irregular performance was actually a fixed and duplicable characteristic of the machine in question. Therefore, they are now firm in their conviction that the claim of 0.1 per cent accuracy is amply justified.

Necessary Modifications to Calculation Technique. To take advantage of the experimental accuracy available, certain modifications of the usual calculating methods have been found to be necessary. On the other hand, a great time and labor saving has been achieved by the elimination of the necessity of correcting the other readings for speed variations.

The fundamental units which measure the performance of any hydraulic machine are the head in feet of the fluid flowing and rate of flow in cubic feet per second of the fluid flowing. These are the units employed by the laboratory and the readings taken must be reduced to these terms.

The pressure gages measure the head in pounds per square inch. These readings are readily converted to head of actual fluid by the use of a conversion factor read from a chart on which is plotted its variation with temperature. However, since the piezometer rings are several pipe diameters away from the machine, and since the velocities may be as high as 40 fps, it is necessary to make a correction for pipe friction. The friction coefficient used is that for normal flow, and any additional friction existing due to nonuniform velocity distribution is charged against the machine.

The procedure for the rate-of-flow measurements is somewhat more complex. The reading of the differential manometer actually is a weight of mercury instead of a height. This helps to eliminate a correction for mercury temperature. By means of a factor from another chart, this reading is converted into feet of water at the temperature of the testing circuit. From

this figure and the temperature, the Reynolds number of the venturi tube in use is graphically determined. Reference to the curve of meter coefficient vs. Reynolds' number gives the correct meter coefficient, and from this and the corrected differential reading the rate of flow is calculated.

The values of torque are correct as read. However, in order to make all the laboratory results comparable, water horsepowers are calculated from the head and rate-of-flow figures by the use of the density of water at standard temperature, that is, 68 F. Therefore, in calculating dynamometer horsepower, it is necessary to reduce the torque figure to the same basis by multiplying it by the ratio of the standard water density to the actual density. The overall efficiency calculations are not affected by this reduction.

Review of Laboratory Facilities. It has been stated that an attempt was made to secure the maximum versatility of laboratory equipment compatible with the known objective. Up to the present time the laboratory has used this equipment only in the study of the problems arising in connection with the design of pumping plants for the Colorado River aqueduct. However, it is obvious that its possibilities of application extend into much wider fields. There are some obvious limitations. High rates of flow at low heads offer difficulties because the line velocities have been kept rather high with correspondingly appreciable friction losses. The dynamometer is essentially a high-speed machine, and below about 2400 rpm the maximum power available decreases directly with speed. Although the gearbox can be set to control any speed below its theoretical maximum of 5555 rpm, it is not feasible to use it below about 1000 rpm because of the increasing load on the gear teeth at the lower rates. Only machines that can operate with horizontal shafts have been provided for, and of course work with open conduits belongs in the hydraulic-structures laboratory rather than in the laboratory herein described. The following classes of problems, however, are within the scope of the equipment, and represent the fields of investigation in which the laboratory will be employed after the completion of its original objectives:

(*a*) Pumps or turbines up to 12 cfs maximum capacity, 700 ft maximum head, 5000 rpm maximum speed, and 400 hp maximum power.

(*b*) Explorations of internal-flow conditions in hydraulic machinery.

(*c*) Valves, conduits, or fittings investigations at high velocities.

(*d*) Similar investigations involving noncorrosive liquids other than water, since system is closed and isolated.

(*e*) In general, problems requiring flows at high Reynolds, numbers and precision of measurements.

Acknowledgments

The primary responsibility for the development of the laboratory and its equipment has rested on the shoulders of the members of the supervising committee named at the beginning of this paper. In addition, special acknowledgment is due to the following men, who have contributed much to the design and construction. Dr. Frank Wattendorf was in charge of the operative staff during the period of construction and initiation of testing. Ralph M. Watson has held the same position for the latter half of the laboratory's existence. Dr. George Wislicenus and Dr. R. C. Binder have carried out much of the detailed design of the mechanical equipment and instruments. Emmet Irwin was responsible for the main electrical design, and E. E. Simmons and H. L. Levinton have both designed and constructed most of the special vacuum-tube circuits. Finally, the successful operation of the special precision instruments and control equipment has been due in a large measure to the exquisite workmanship of Fred C. Henson, in whose local instrument shop most of them were constructed.

The Effect of Installation on the Coefficients of Venturi Meters

By W. S. Pardoe,[1] Philadelphia, Pa.

The author presents the results of a series of tests conducted to determine the effect of the location of a venturi meter on its coefficient of discharge. These results are shown graphically together with sketches of the type of installations and venturi meters for which the coefficients were determined. The author shows mathematically and experimentally that for normal turbulence and shooting flow the coefficients are the same for low-ratio meters, but that for high-ratio meters the coefficient increases from that for shooting flow to the normal value in about 20 diameters of mixing lengths.

The figures in this paper show the results of tests on three venturi-meters of the Herschel type made by the Simplex Valve and Meter Company, Philadelphia, Pa. One of these was a cast-iron, heavy-pressure 11.956 × 8.2285-in. meter with a ratio of 0.688. The second was a 125-lb bronze standard 8.060 × 3.3759-in. meter with a ratio of 0.419. The third was a cast-steel, heavy-pressure 7.810 × 5.0046-in. meter with a ratio of 0.640. All three meters have a $10^1/_2$-deg upstream half angle and $6^1/_2$-deg downstream half angle.

The tests were conducted at the hydraulic laboratory of the civil engineering department of the University of Pennsylvania which is shown schematically in Fig. 1. It is composed of a pumping sump of 30,000-gal capacity; five centrifugal pumps with a total capacity of 12 cfs; a system of 8 to 12-in. cast-iron pipes; a pressure tank 40 ft high, 5 ft 6 in. in diameter, with 6-in. and 24-in. nozzles and two 16-in. overflows located 10 ft and $22^1/_2$ ft, respectively, above the 6-in. nozzle; a 12-in. standpipe in parallel with the pressure tank with overflows at $2^1/_2$-ft and 5-ft intervals to a point 50 ft above the 6-in. nozzle for the purpose of taking the excess water pumped above that used, thus maintaining a constant head on any piece of apparatus in the laboratory; a set of 8-in. and 12-in. wrought-iron pipe lines in which were placed the venturi meters being tested and at the ends of which were placed control valves with the discharge ends cut off to prevent short-tube action; and two 16,000-lb weighing tanks with a capacity of 10 cfs. The scales used with the weighing tanks are sensitive to 1 lb and the beam moves throughout its range for 5 lb.

During the tests, time was taken with two stop watches calibrated in terms of a seconds pendulum.

Fig. 2 shows the differential gages used in the tests. Heads from 40 to 5 ft are measured on a direct-reading mercurial differential gage with an inside diameter of $^3/_4$ in. and with a vulcanite float on top of the mercury. Heads from $4^1/_2$ to 1 ft are read on a direct-reading air-differential pot gage the tube of which has an inside diameter of $^3/_4$ in.; a short tube in parallel with the pot gives similar menisci. Heads of less than 1 ft are measured on a 3-in. air-differential hook gage capable of being read to 0.001 ft and of interpolation to 0.0001 ft. The reading of the air-differential gage was corrected for the weight of the air by using the equation

$$h = y[1 - (w'/w)]$$

where h = venturi head, ft; y = gage reading, ft; w' = weight of air at main pressure, lb per cu ft; and w = weight of water, lb per cu ft.

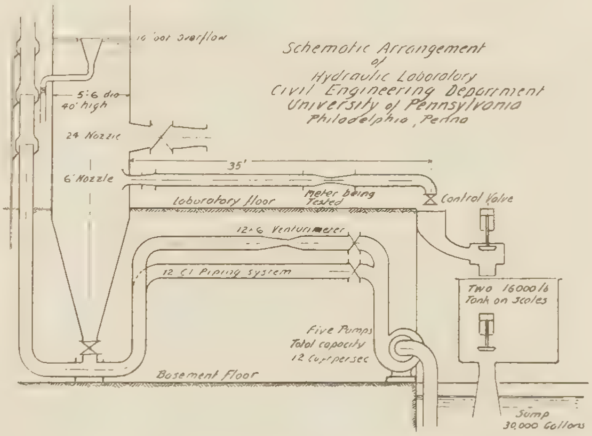

Fig. 1 Schematic Arrangement of the Hydraulic Laboratory at the University of Pennsylvania

Fig. 4 is a summary of Figs. 5 to 14, inclusive, and shows results of three meters placed in various types of flow from normal turbulence to shooting flow. Results of normal-turbulence conditions are shown in Figs. 5, 9, and 12, while results of shooting-flow conditions are shown in Figs. 8, 11, and 14. Velocity traverses taken with a plain pitot tube are shown for the 8-in. meters. These curves show that the coefficient for normal turbulence and shooting flow are the same for the low-ratio meters, but that for the high-ratio meters it increases from that for shooting flow to the normal value in about 20 diameters of mixing length. This can be shown mathematically as follows.

The ordinary expression for the venturi meter may be written

$$V_2 = \frac{C}{\sqrt{1 - (d_2/d_1)^4}} \sqrt{2g(P_1 - P_2)} \quad \ldots\ldots\ldots [1]$$

If the frictional term $k(V_2^2/2g)$ be included in Bernoulli's theorem and a factor α be used to allow for the excess of kinetic

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[1] Professor of Hydraulic Engineering, University of Pennsylvania. Professor Pardoe was graduated in mechanical engineering from the Ontario School of Practical Science and from the University of Toronto in applied science. He spent three years at marine-engine and pump drafting and was employed as hydraulic engineer for three years by the Canada Foundry Company. He has served for twenty-seven years as instructor, assistant professor, and professor of hydraulic engineering at the University of Pennsylvania, and as consulting engineer on hydraulic power, dams, and hydraulic problems.

Contributed by the Special Research Committee on Fluid Meters and the Hydraulic Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held at New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

power in the main over the kinetic power computed from the mean velocity V_1 the equation becomes

$$Z_1 + P_1 + \alpha \frac{V_1^2}{2g} = Z_2 + P_2 + \frac{V_2^2}{2g} + k \frac{V_2^2}{2g} \quad [2]$$

or

$$V_2 = \frac{1}{\sqrt{1 - \alpha(d_2/d_1)^4 + k}} \sqrt{2g(P_1 - P_2)} \quad [3]$$

Equating Equations [1] and [3] it is seen that

$$C = \sqrt{\frac{1 - (d_2/d_1)^4}{1 - \alpha(d_2/d_1)^4 + k}} \quad [4]$$

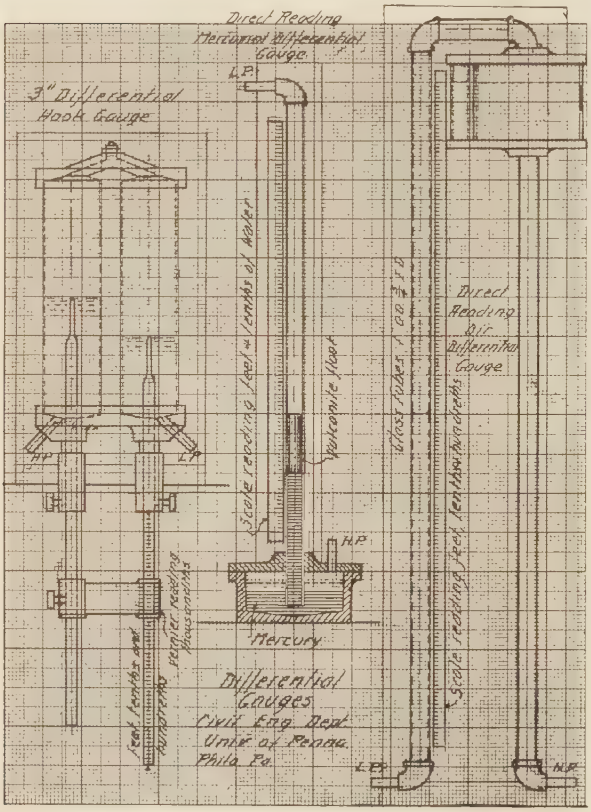

FIG. 2 DIFFERENTIAL GAGES

To compute the value of α, assume that the velocity traverse is an ellipse, as shown in Fig. 3, with a wall velocity equal to one half the central velocity. The equation of the ellipse is

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$$

where

$$x = \frac{a}{b}\sqrt{b^2 - y^2}$$

With reference to Fig. 3

$$x = \frac{V_c}{2r}\sqrt{r^2 - y^2} \quad [5]$$

$$\text{Kinetic power} = W\frac{V^2}{2g} = wa\frac{V^3}{2g} \quad [6]$$

$$V = \frac{V_c}{2} + \frac{V_c}{2r}\sqrt{r^2 - y^2} \quad [7]$$

$$da = 2\pi y dy$$

Therefore, the kinetic power, hereafter denoted as K, can be expressed as

$$K = 2\pi\left(\frac{V_c}{2}\right)^3\left[1 + \frac{1}{r}\sqrt{r^2 - y^2}\right]^3 y dy \frac{w}{2g} \quad [8]$$

$$= \int_0^r 2\pi\left(\frac{V_c}{2}\right)^3\left[y + \frac{3y}{r}(r^2 - y^2)^{1/2} + \frac{3y}{r^2}(r^2 - y^2) + \frac{y}{r^3}(r^2 - y^2)^{3/2}\right] dy \frac{w}{2g}$$

$$= \frac{\pi V_c^3}{4}\left[\frac{y^2}{2} - \frac{(r^2 - y^2)^{3/2}}{r} - \frac{3}{4}\frac{(r^2 - y^2)^2}{r^2} - \frac{(r^2 - y^2)^{5/2}}{5r^3}\right]_0^r \frac{w}{2g}$$

$$= \frac{\pi V_c^3}{4}\left[\frac{r^2}{2} + r^2 + \frac{3}{4}r^2 + \frac{1}{5}r^2\right]\frac{w}{2g}$$

$$= 0.6125\, \pi V_c^3 r^2 \frac{w}{2g} \quad [9]$$

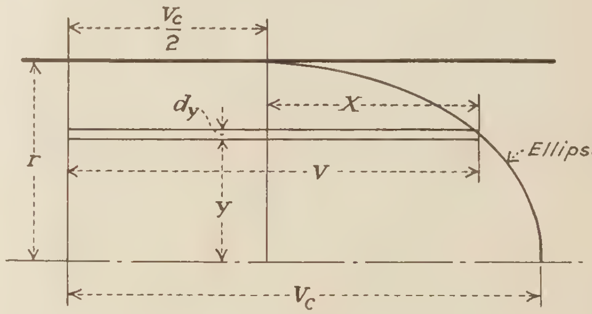

FIG. 3

The kinetic power for the average velocity is as follows as the mean velocity can be expressed by the equation

$$\frac{V_c}{2} + \frac{2}{3}\frac{V_c}{2} = \frac{5}{6}V_c \quad [10]$$

from which

$$K = w\pi r^2\left(\frac{5}{6}V_c\right)^3 \Big/ 2g \quad [11]$$

and

$$\alpha = 0.6125 \Big/ \left(\frac{5}{6}\right)^3 = 1.0584 \quad [12]$$

If all corrections be made to Bernoulli's theorem and assuming shooting flow at the throat

$$Z_1 + P_1 + 1.0584\frac{V_1^2}{2g} = Z_2 + P_2 + \frac{V_2^2}{2g} + k\frac{V_2^2}{2g} \quad [13]$$

or

$$V_2 = \frac{1}{\sqrt{1 - 1.0584\,(d_2/d_1)^4 + k}}\sqrt{2g\,(P_1 - P_2)} \quad [14]$$

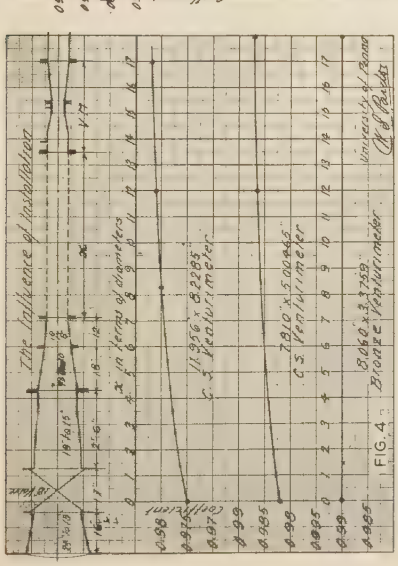

FIG. 4

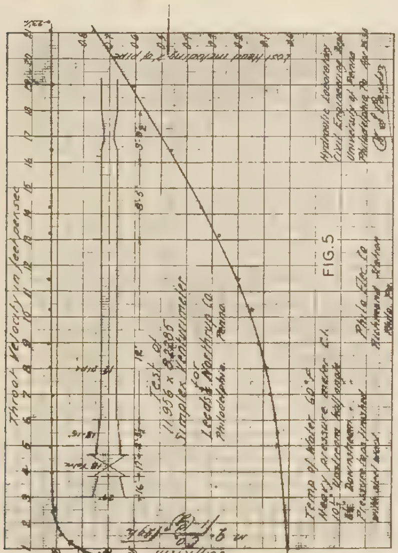

FIG. 5

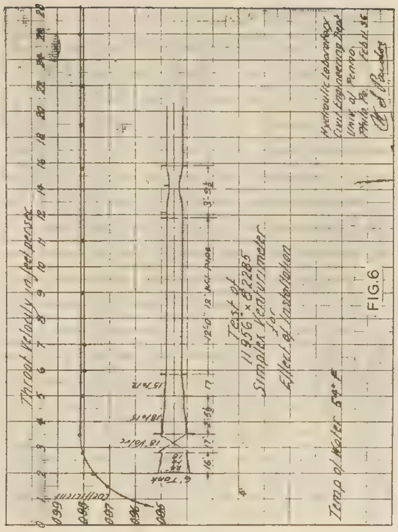

FIG. 6

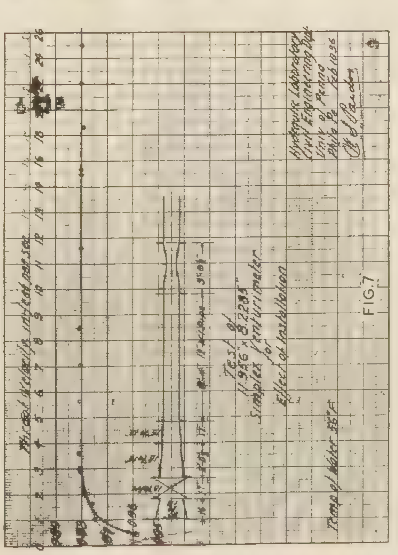

FIG. 7

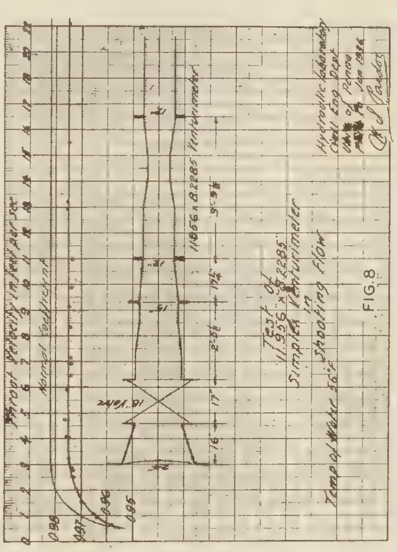

FIG. 8

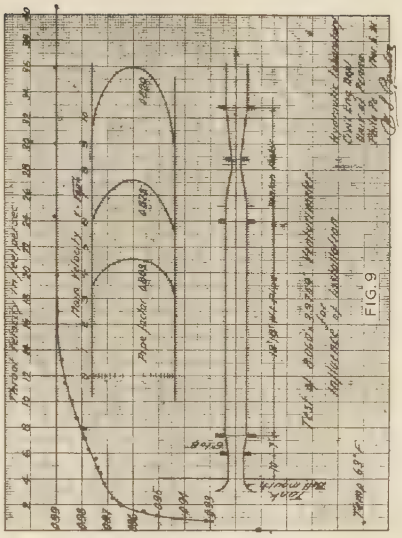

FIG. 9

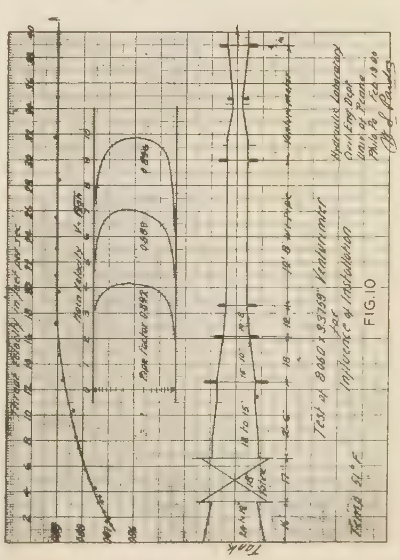

FIG. 10

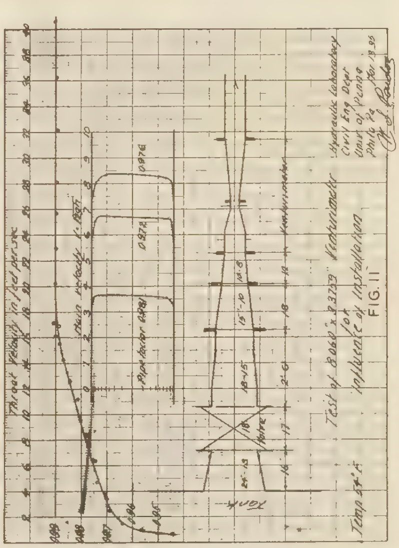

FIG. 11

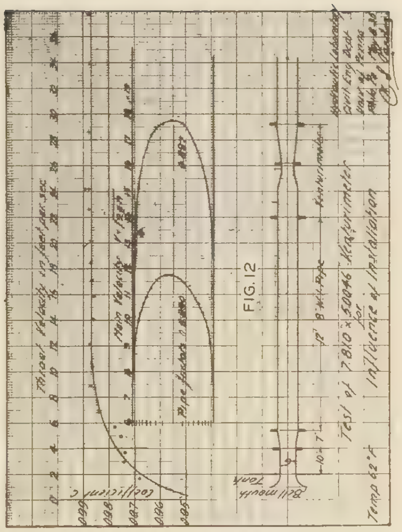

FIG. 12

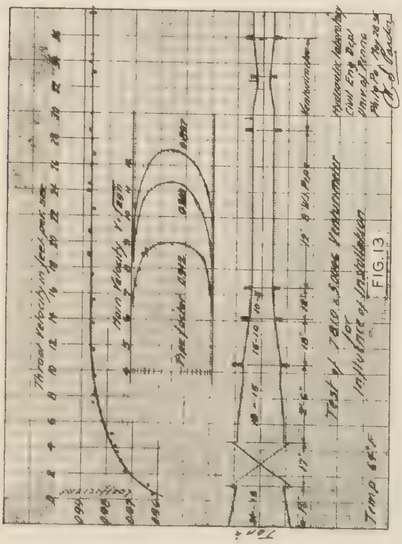

FIG. 13

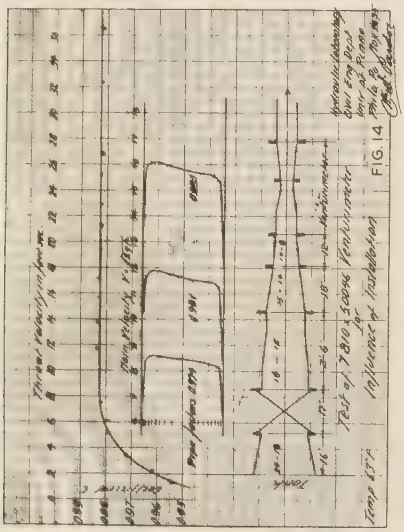

FIG. 14

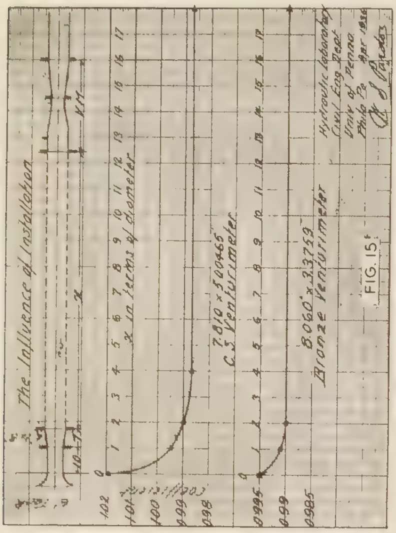

FIG. 15

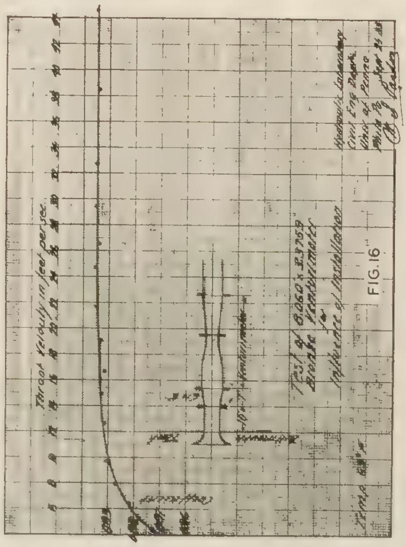

FIG. 16

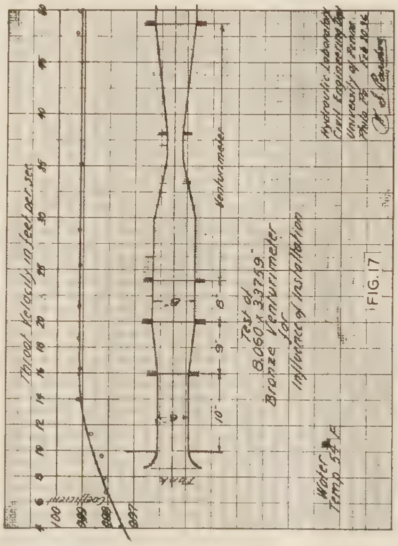

FIG. 17

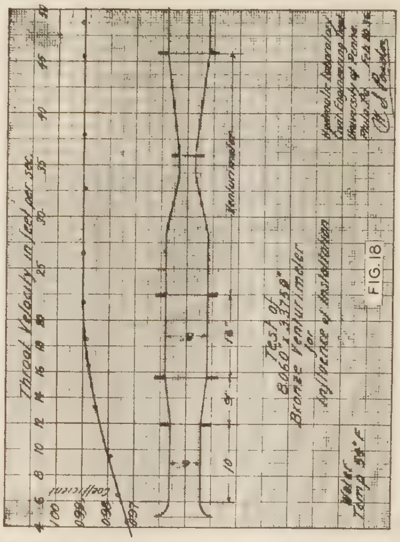

FIG. 18

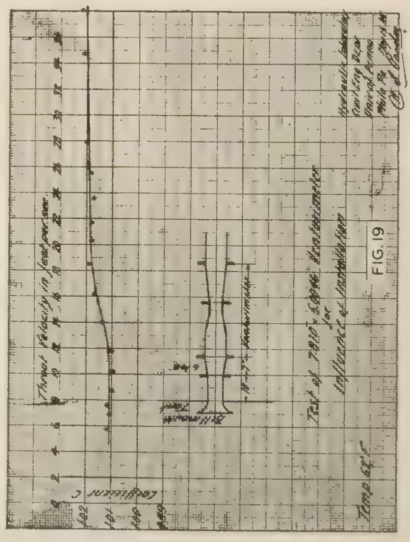

FIG. 19

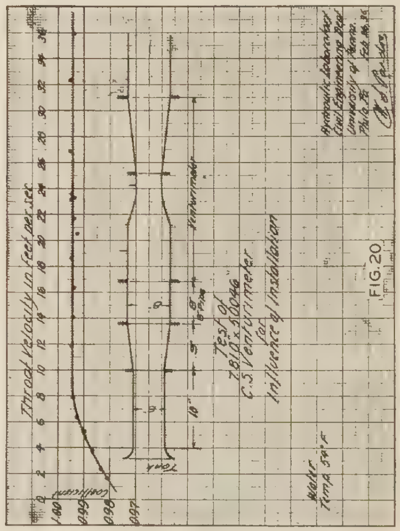

FIG. 20

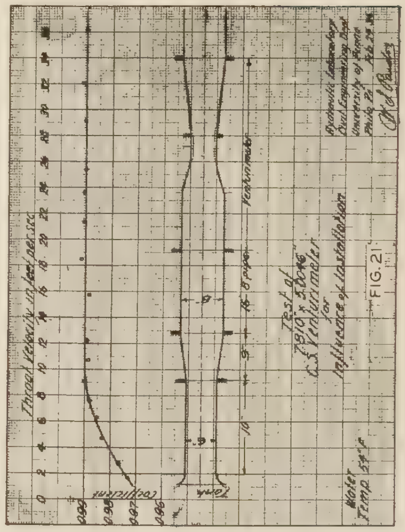

FIG. 21

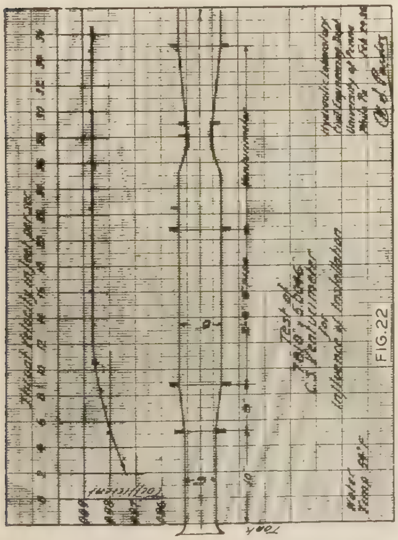

FIG.22

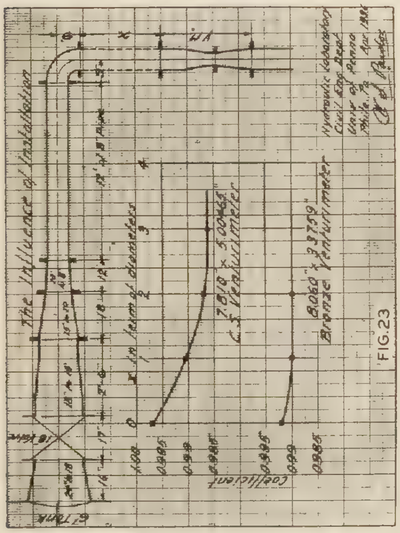

FIG.23

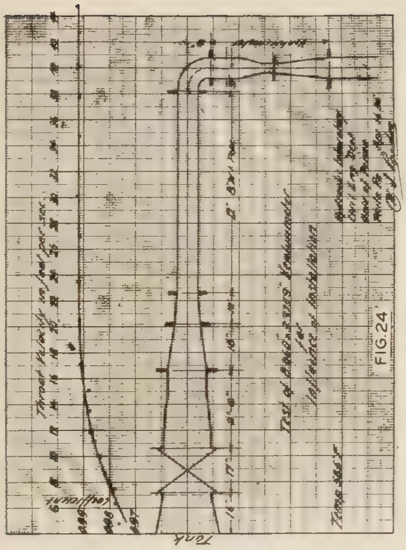

FIG.24

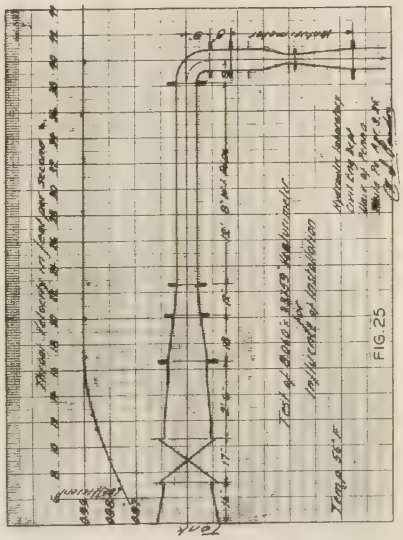

FIG.25

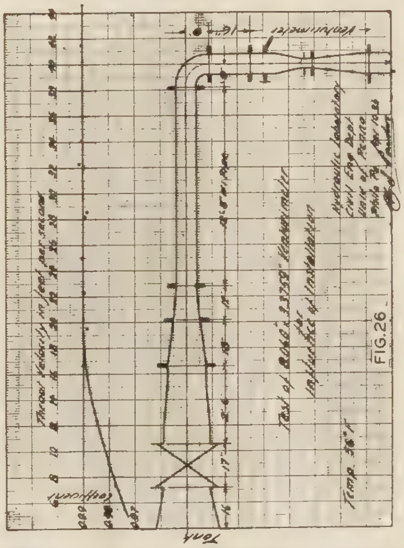

FIG.26

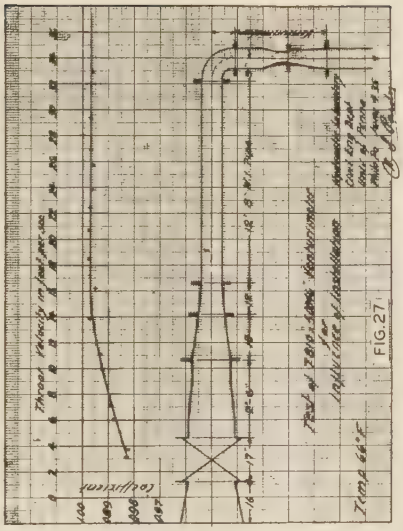

FIG.27

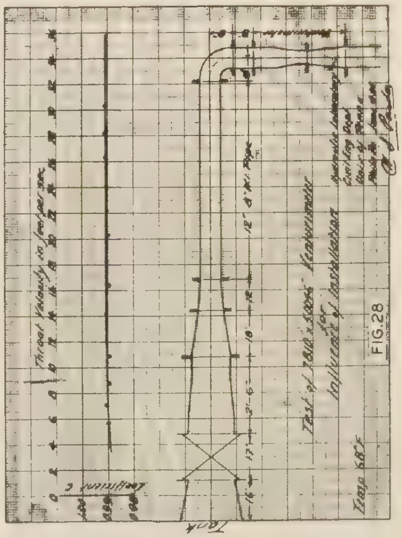

FIG.28

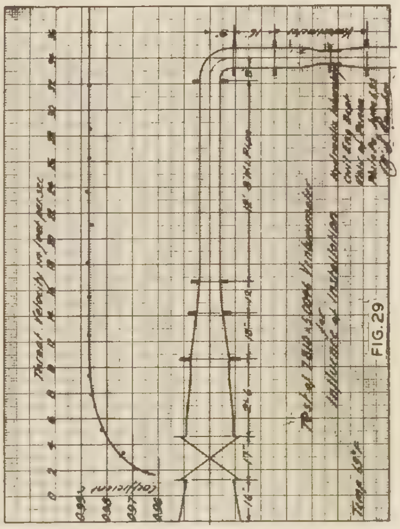

FIG.29

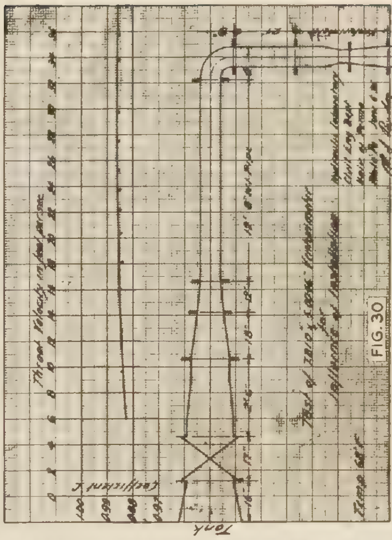

FIG.30

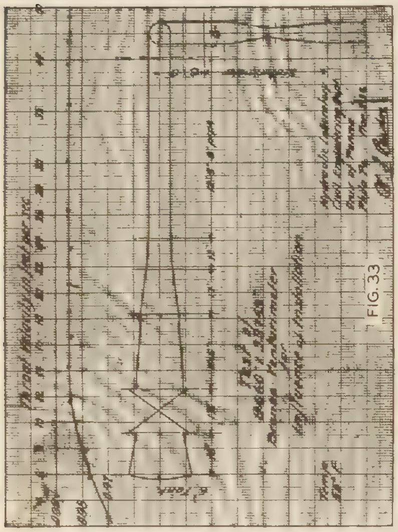
FIG. 33

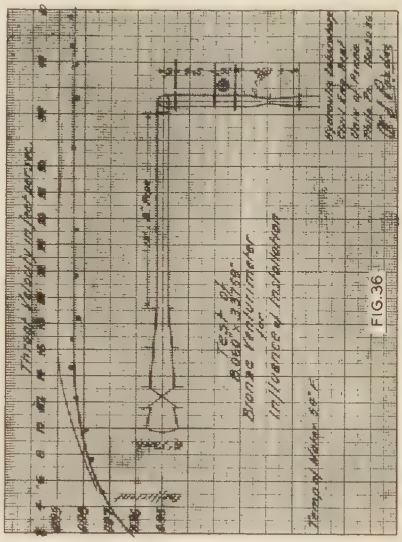
FIG. 36

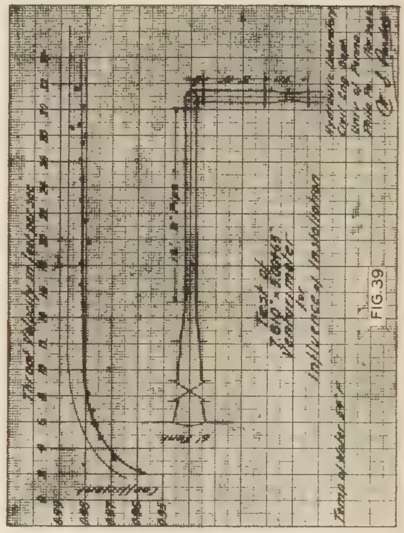
FIG. 39

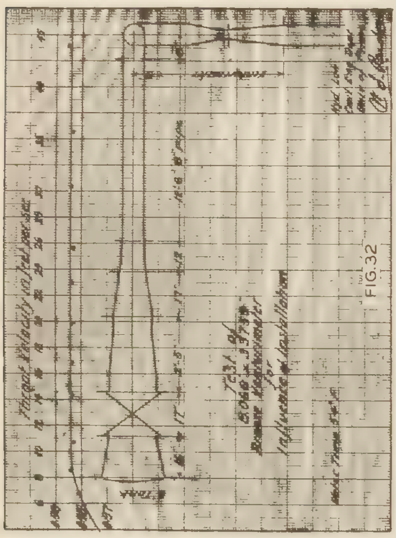
FIG. 32

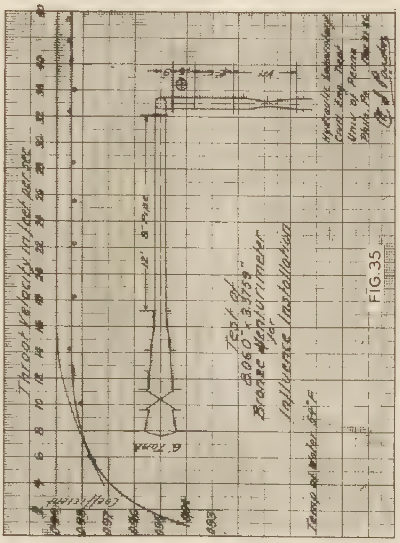
FIG. 35

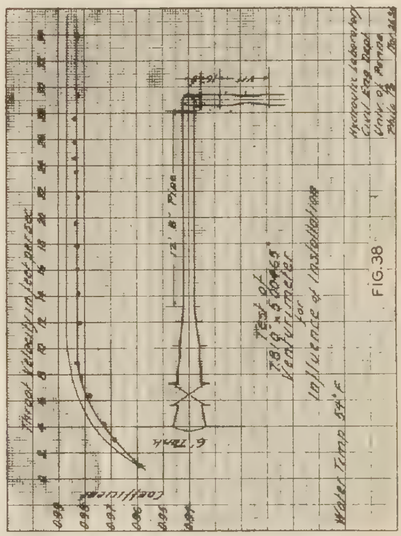
FIG. 38

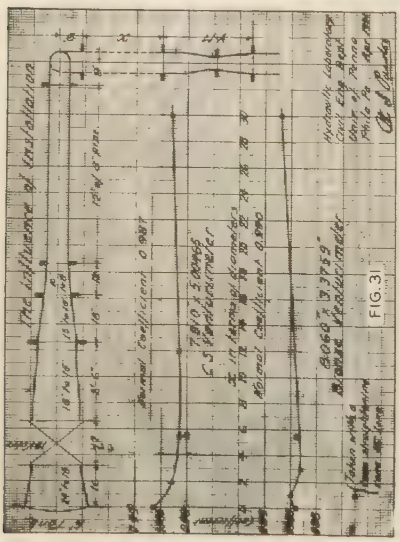
FIG. 31

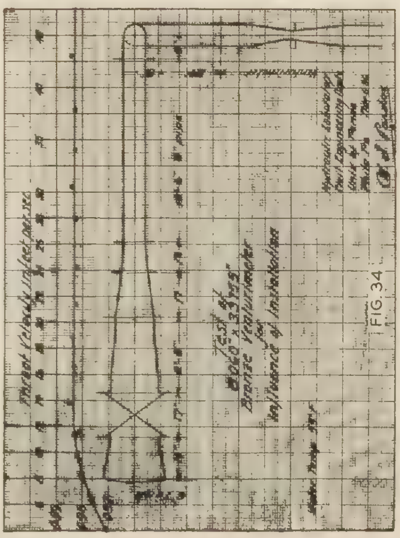
FIG. 34

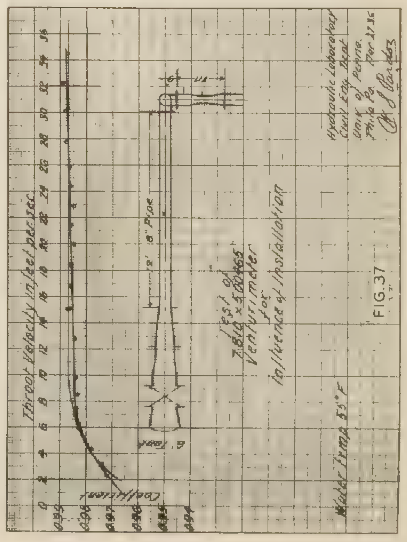
FIG. 37

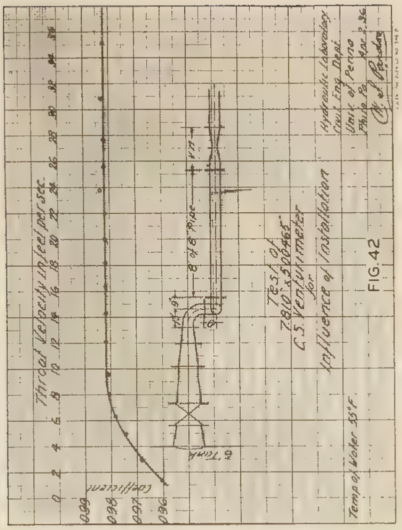

FIG. 42

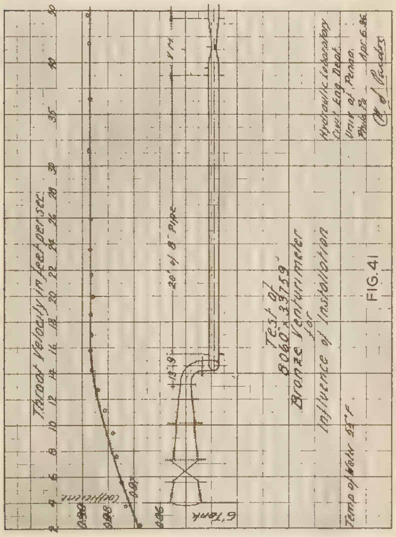

FIG. 41

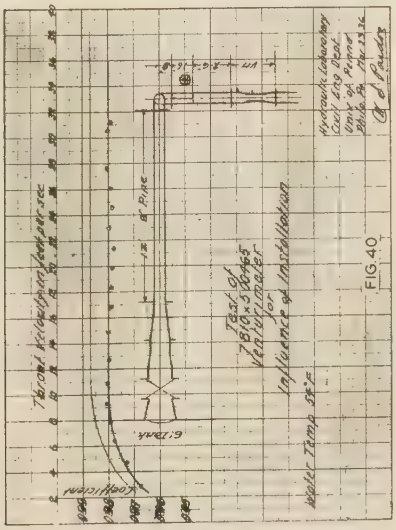

FIG. 40

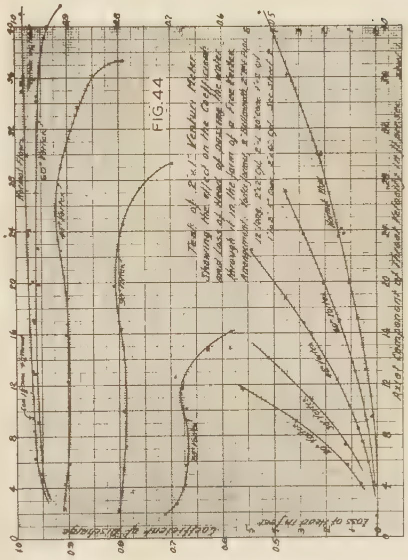

FIG. 44

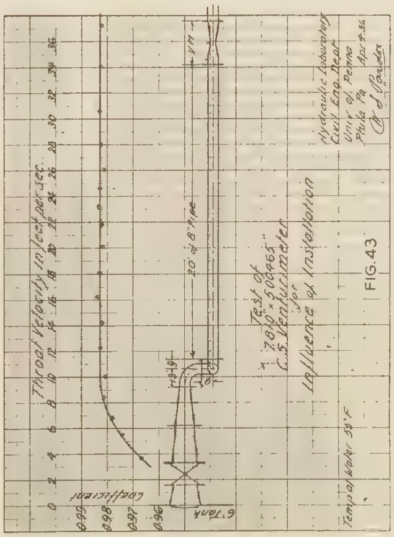

FIG. 43

also

$$V_2 = \frac{C}{\sqrt{1 - (d_2/d_1)^4}} \sqrt{2g(P_1 - P_2)} \cdots\cdots [15]$$

Therefore

$$C = \sqrt{\frac{1 - (d_2/d_1)^4}{1 - 1.0584\,(d_2/d_1)^4 + k}} \cdots\cdots\cdots\cdots [16]$$

and

$$k = \frac{1 - (d_2/d_1)^4}{C^2} - \left[1 - 1.0584\left(\frac{d_2}{d_1}\right)^4\right] \cdots\cdots [17]$$

The coefficient for shooting flow C_s may be written

$$C_s = \sqrt{\frac{1 - (d_2/d_1)^4}{1 - (d_2/d_1)^4 + k}} = \sqrt{\frac{1 - (d_2/d_1)^4}{1 - (d_2/d_1)^4 + \frac{1 - (d_2/d_1)^4}{C^2} - \{1 - 1.0584(d_2/d_1)^4\}}} \cdots\cdots [18]$$

since $\alpha = 1.0$.

The foregoing theory was checked on the three meters, the coefficients of which are shown in Figs. 5 and 8, Figs. 9 and 11, and Figs. 12 and 14, and accurate results given in Table 1 were obtained.

TABLE 1 COMPARISON OF CALCULATED AND OBSERVED VENTURI-METER COEFFICIENTS

Size of meter, in.	Coefficients — Experimental C	Experimental C_s	Computed C_s
11.956 × 8.2285	0.982	0.975	0.974
8.060 × 3.3759	0.990	0.990	0.989
7.810 × 5.0046	0.987	0.982	0.981

In each case the computed value was 0.1 per cent below the experimental value. This may be the result of the assumption of a low traverse coefficient or pipe factor of 0.833.

The theory and results emphasize the fact that a high traverse coefficient or pipe factor produces a low coefficient, and a low pipe factor a high coefficient.

Fig. 15 is a summary of Figs. 9, 12, and 16 to 22, inclusive, in which the high- and low-ratio 8-in. meters were placed at varying distances below the 6-in. nozzle and 6 to 8-in. flanged fitting. This simulates the conditions of a centrifugal pump, which is discharging the water without a whirl. The low-ratio meter attains its normal coefficient of 0.99 in two diameters. The high-ratio meter requires four diameters to reach its normal coefficient of 0.987. These approach conditions produce a low pipe factor and high coefficient.

Fig. 23 is a summary of Figs. 24 to 30, inclusive, in which the high-ratio and low-ratio 8-in. meters were placed at various distances from a 90-deg short-radius 8-in. elbow. The high-ratio meter required three diameters and the low-ratio meter one diameter to reach their normal coefficients of 0.987 and 0.990, respectively. The elbow produces a high velocity at the outside; immediately following it, a low pipe factor and a high coefficient.

Fig. 31 is a summary of Figs. 32 to 43, inclusive, in which the high- and low-ratio meters are placed at varying distances after two 90 deg short-radius elbows in planes at 90 deg to each other. This apparently produces a free vortex in the meter and lowers the coefficient below normal. Up to x equals four diameters, there seems to be the opposing effects of low pipe factor tending to raise the coefficient and the free vortex tending to lower it. After four diameters the coefficient rose, but it did not reach the normal coefficients in any feasible length. Cross straightening vanes, two diameters long, were not effective in restoring the coefficients to normal.

Fig. 44 shows the effect on the coefficient of a 2 × 1-in. bronze meter of passing the water through it in the form of a free vortex. The coefficients for all angles of whirl are lowered.

Acknowledgments are due to the Simplex Valve and Meter Company for supplying the venturi meters, to E. F. Stover, Instructor in Civil Engineering, University of Pennsylvania, and the following graduate students: L. H. Glassman, H. J. Koch, and H. F. Watson, who assisted on some of the tests.

A Study of Cutting Fluids Applied to the Turning of Monel Metal

By O. W. BOSTON[1] and W. W. GILBERT,[2] ANN ARBOR, MICH.

This paper presents the results of an experimental study of the application of cutting fluids to the turning of monel metal. The manner in which high-speed-steel tools wear and fail is discussed. By competitive tests the tool life of the large-radius tool is compared with that of a straight-edged small-radius tool. The practical advantages of each are discussed. Cutting-speed tool-life curves are obtained and equations developed for the latter tool when turning monel metal with deep thin cuts and shallow thick cuts for each of several types of cutting fluids. The influence of feed and depth of cut on tool life at various cutting speeds is determined when turning monel metal with a sulphurized mineral oil. Finally the three components of the cutting force are determined for various combinations of feed and depth of cut when turning monel metal with an emulsion.

DATA dealing with the subject of tool life and cutting forces when turning monel metal with high-speed-steel tools have been obtained which show the influence of the type of cutting fluid involved. This represents a progress report of experiments conducted in the department of metal processing at the University of Michigan.

The monel metal was furnished as three bars, each $7^3/_{16}$ in. diameter by 20 in. long. It had been forged and rough-turned and was of the regular temper having the following analysis in percentages: 0.12 C, 0.005 S, 0.07 Si, 0.81 Mn, 1.16 Fe, 31.21 Cu, and 66.49 Ni. In transverse test these bars were found to have a tensile strength of 95,000 lb per sq in., a yield point of 78,000 lb per sq in., a proof stress of 62,500 lb per sq in., an elongation in 2 in. of 23 per cent; and a reduction in area of 55.5 per cent. The Brinell hardness values, using the 3000-kg load, from the center to the surface ranged from 207 to 224. This rather wide range led to considerable variation in test results. One of the bars particularly proved to be quite different from the other two in its machinability rating. For this reason tool-life tests had to be run at cutting speeds which would give short values of tool life in order to conserve material.

The equipment used in these tests consisted of a heavy-duty engine lathe with a 14-in. swing, a 6-ft bed, and a 16-speed geared-head drive with a variable-speed direct-current motor provided with a 66-point rheostat as described in another paper.[3] The cutting-fluid pump circulated 4.75 gpm.

The cutting tools consisted of $^3/_8$-in.-square standard high-speed steel tool bits of the following analysis in per cent: 0.77 C, 0.28 Si, 0.24 Mn, trace S, 0.020 P, 17.64 W, 4.09 Cr, and 1.38 Va. The bits were selected from the same heat and were handled through the hardening furnaces on trays, there being 12 tool bits to each tray load. They were preheated to 1620 F for 15 min, then transferred to a high-temperature furnace of a semimuffle type at 2360 F for 3 min. The atmosphere in the high-temperature furnace contained approximately 1.5 per cent carbon monoxide with no oxygen present. Following an oil quench, the bits were drawn in a Homo electric furnace at 1060 F for 1.5 hr.

The tools were machine-ground first on a 46-grit *B* bond vitrified wheel, and then finish-ground on a No. 38100GS Norton Company wheel. This produced a surface finish much superior to that of the ordinary lathe tool. The radius of each tool was ground on a specially constructed machine to conform to the contour of a special cam. The tools used in these tests were ground to the shape of 8-22-6-6-6-15-$^3/_{64}R$, that is, an 8-deg back rake angle, a 22-deg side rake angle, a 6-deg side clearance angle, a 6-deg end clearance angle, a 6-deg end cutting-edge angle, a 15-deg side cutting-edge angle, and a $^3/_{64}$-in. nose radius. From a group of tools so prepared a number was selected for uniformity by Rockwell hardness tests so that they were within the Rockwell *C* scale of 65 to 67. Frequently some of these selected tools would produce tool-life values consistently high or low and were, therefore, discarded.

In running the tool-life tests on monel metal with high-speed-steel tools it was first necessary to determine the type of tool failure and the exact time of its occurrence. Two views of a tool after having failed are shown in Fig. 1. The upper view shows the face of the tool in which the crater has been worn. The land supporting the built-up edge is seen just back of the cutting edge. Grinding marks on the land are seen to be continuous with those on the face of the other side of the crater, indicating a more or less permanent main portion of the built-up edge. The lower illustration shows the flank of the tool below the active cutting edge. The small narrow ridge is a portion of the flank which has been abraded. The V-shaped notch at the tool point shows how the flank below the nose is abraded when the tool fails. It is seen that the tool wears progressively by cupping on the face and abrading on the flank. The deep abrasion

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[2] Instructor, College of Engineering, University of Michigan, Jun. A.S.M.E. Dr. Gilbert was graduated in mechanical engineering from the University of Colorado in 1931, received his master's degree in 1932, and his Sc.D. degree in 1935 from the University of Michigan. Since January, 1932, Dr. Gilbert has worked with Professor Boston on various phases of metal cutting and is coauthor of several metal-cutting papers.

Contributed by the Machine Shop Practice Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held at New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

[3] "The Influence of Cutting Fluids on Tool Life in Turning Steel," by O. W. Boston, W. W. Gilbert, and C. E. Kraus, Trans. A.S.M.E., vol. 58, July, 1936, paper RP-58-11, p. 371.

below the nose is caused quite suddenly at the instant of tool breakdown. The instant that this occurs is manifested by a change in the machine finish on the work. The subsequent finish is slightly rougher and the work is slightly oversized in diameter. Cutting will continue with apparent satisfaction,

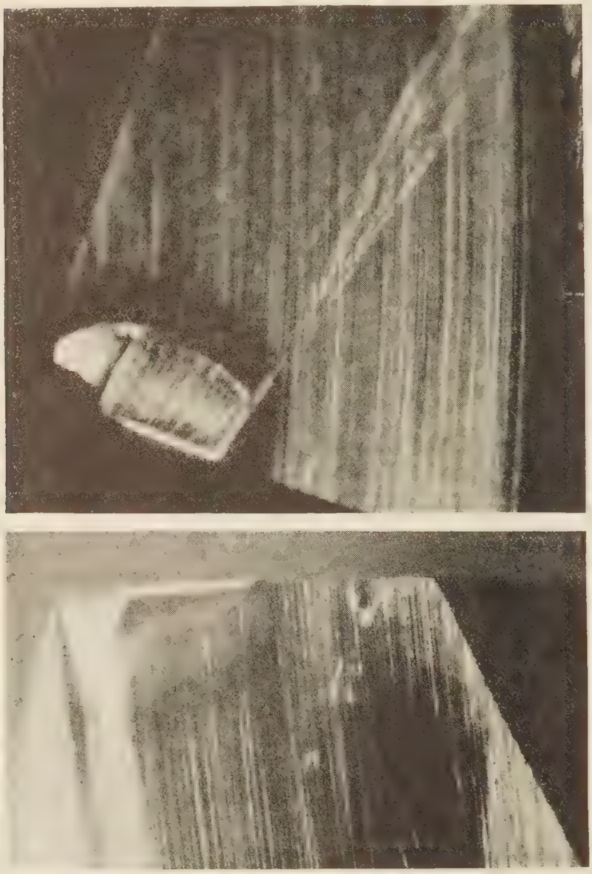

Fig. 1 Types of Tool Failure After Cutting Monel Metal

(The face and flank of a high-speed-steel tool which has failed by cupping in turning monel metal. Depth of cut = 0.100 in., feed = 0.0127 in. per revolution, width of land supporting built-up edge = 0.0044 in., the width of crater from the land = 0.05 in., and the width of the land worn on flank = 0.004 in.) ×12

however, for some appreciable time after this point of breakdown is reached.

As a newly ground tool starts to cut, the chip is removed in a long rather straight ribbon. As cupping progresses the chips become coiled helically. These helixes are also continuous and change only in diameter as the tool wear progresses. The under side of the chip appears to be highly burnished, indicating the presence of a very small built-up edge. The machined surface also appears to be highly burnished and lacks the customary torn appearance present when turning steel.

Influence of Tool Shape

The authors have used a tool shape *A* shown in Fig. 2 in many of their investigations in turning steel. For the sake of comparison it was desirable to continue the use of this tool in the experimental work on monel metal. The manufacturers of monel metal recommend several shapes of tools, one of which closely approximates that described previously. Their tool is shown as tool *B* in Fig. 2. Tool *B* differs from tool *A* in that it has a 15-deg instead of a 22-deg side rake angle, a $^1/_8$-in. instead of a $^3/_{64}$-in. nose radius, and a 10-deg instead of a 6-deg end clearance angle. Tool-life tests turning dry were run with the two tool shapes for two sizes of cut as shown in Fig. 2. For the 0.100-in. deep cut and 0.0127-in. feed practically identical values were obtained. Tool *B* has, if anything, a slight preference for this size of cut. For the shallow thick cut of 0.050 in. depth and 0.0255 in. feed,

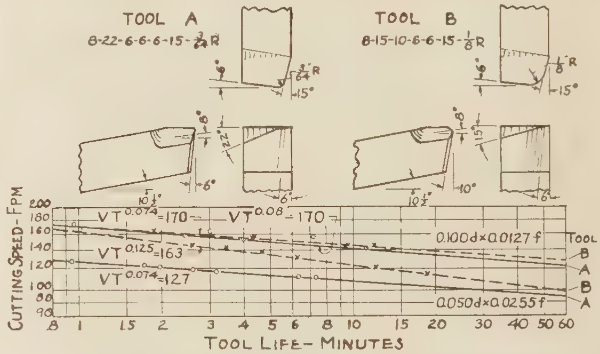

Fig. 2 Cutting-Speed Tool-Life Curves for Dry Turning Monel Metal, Comparing Two Tool Shapes

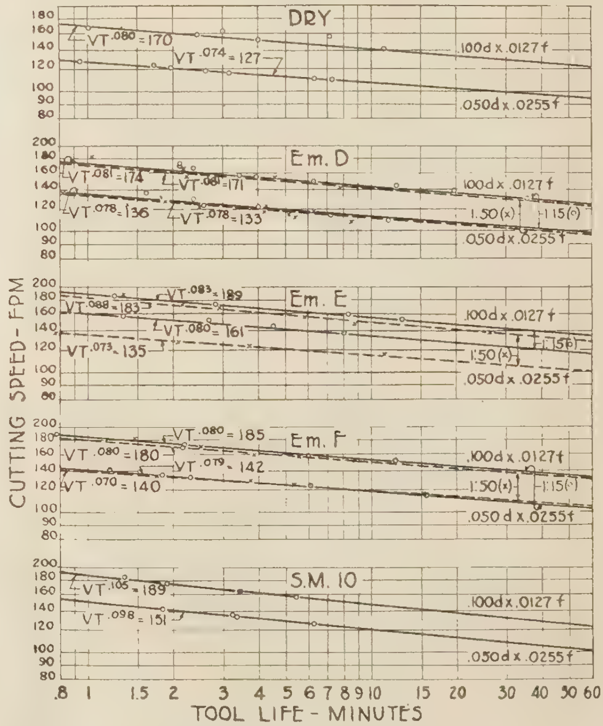

Fig. 3 Cutting-Speed Tool-Life Curves

(High-speed-steel tools 8-22-6-6-6-15-$^3/_{64}R$ turning monel metal using five types of cutting fluids for each of two sizes of cut. Em. *D* is an emulsion of an acid-sludge soluble oil; Em. *E* is a transparent emulsion; Em. *F* is a milky emulsion of a soap-base soluble oil; and *S.M.* 10 is a sulphurized mineral oil. The figures 1:15 and 1:50 represent 1 part oil to 15 and 50 parts of water, respectively. The letters *d* and *f* represent depth of cut and feed, respectively.)

tool *B* is superior to tool *A* for values of tool life less than 120 min. Because of the greater steepness of the tool-*B* curve, however, it becomes inferior to tool *A* for values of tool life above 120 min. It was observed that chips produced by tool *B* having a larger nose radius and less rake were coiled irregularly into short helical spirals of small diameter. For breaking up chips, tool *B*

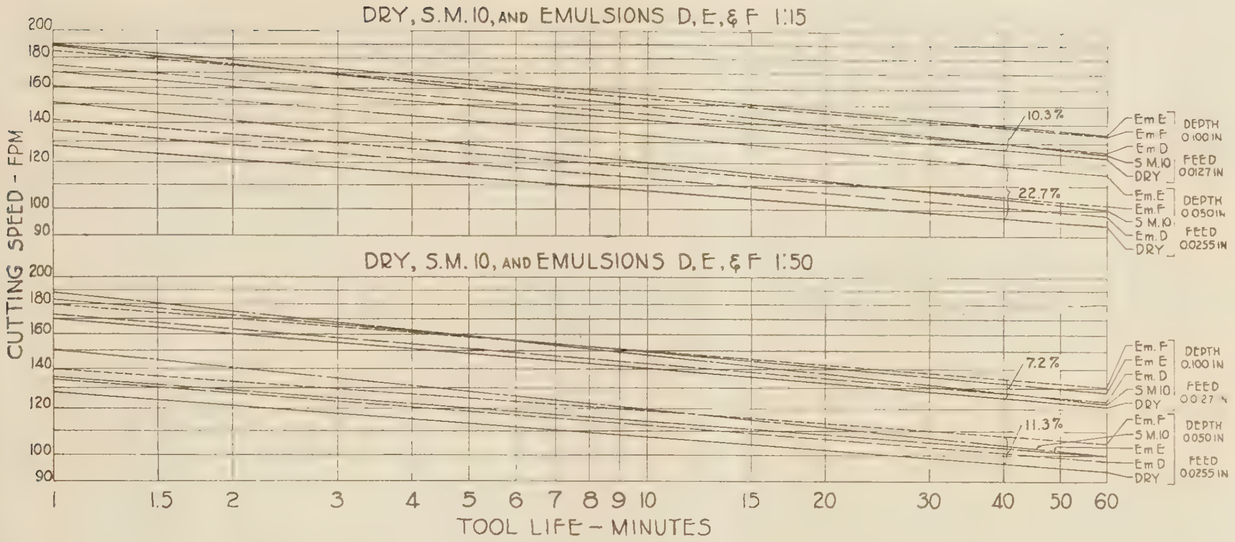

Fig. 4 Summary of Cutting-Speed Tool-Life Curves of Fig. 3 to Show Relative Value of Cutting Fluids for Each Size of Cut

would be recommended except that the chips were forced back against the machined surface, scratching the smooth finish produced by the cutting tool.

Influence of Cutting Fluids

A series of cutting-speed tool-life curves were obtained in which five cutting fluids were used, each with a deep thin cut and a shallow thick cut of the same cross-sectional area. The results are shown graphically on log-log coordinates in Fig. 3. The two curves at the top show the results for dry cutting. The curve for the 0.100 in. depth by 0.0127 in. feed is shown to be well above the curve for the 0.050 in. depth by 0.0255 in. feed. The relative height of these curves is indicated by the value of the constant shown. This constant represents the cutting speed for a tool-life value of 1 min. The two curves are practically parallel. Their vertical displacement, however, shows that the cutting speed for any value of tool life is approximately 33 per cent higher for the deep thin chip.

The curves under the heading of Em. *D* were obtained when a soluble oil of an acid-sludge type was made into an emulsion. One mixture of emulsion was made of 1 part soluble oil to 15 parts of water, and a second was made with 1 part soluble oil to 50 parts water, to represent rich and lean emulsions, respectively. Cutting-speed tool-life curves were obtained for each of these emulsions for each of two sizes of cuts. The curves under Em. *D* show that the richer emulsion gave slightly higher cutting speeds for any tool life. The curves also show that the cutting-speed values for a given tool life will be approximately 28 per cent higher for the deep thin cut.

The Em. *E* curves also show that the values of cutting speed for any tool life are higher for the richer mixtures of emulsion and also that those values for the deep thin cuts are well above those for the shallow thick cuts. The 1-to-15 emulsion of soluble oil *E* materially increases the cutting-speed values for any tool life over those of the lean mixture for the shallow thick cuts of 0.050 × 0.0255 in.

The emulsions of soluble oil *F* also show the cutting-speed curves for the deep thin cuts to be higher than those of the shallow thick cuts. The richer emulsion gives cutting speeds for a given tool life slightly above those of the leaner mixture for the deep thin cut. The cutting-speed curves of the two mixtures for the shallow thick cut are practically identical, but cross at a tool-life value of approximately 5 min. For greater values of tool life, the cutting speeds are higher for the leaner mixture. Using the emulsion Em. *F* it is seen that the cutting speeds for the deep thin cuts are about 28.5 per cent above those of the shallow thick cuts.

The lowest set of curves of Fig. 3 shows values of cutting speed for various values of tool life when taking two sizes of cut with a sulphurized mineral oil. The cutting-speed values for the deep thin cut are about 25 per cent higher than for those of the shallow thick cuts.

All of the curves shown in Fig. 3 appear to be practically parallel. They have been extended to include a tool life up to 60 min. The formula for each curve expressing the relationship between the cutting speed and tool life is shown. The exponent of T, tool life, varies from 0.074 to 0.080 when cutting dry, from 0.098 to 0.105 for *S.M.* 10, from 0.070 to 0.080 for emulsion Em.*F*, and around 0.080 for the emulsion Em. *D*.

The curves of Fig. 3 are grouped according to the cutting fluid used. These curves are again shown in Fig. 4, but grouped according to the richness of the emulsions for each of the two sizes of cut. The maximum increase in cutting speed for a 40-min. tool life when using the rich 1-to-15 emulsion on the 0.100-in. cut at 0.0127 in. feed is shown to be from 126 to 139 fpm, or 10.3 per cent. Similarly for the rich emulsion on the 0.050-in. cut at 0.0255 in. feed, the increase is from 97 to 119 fpm, or 22.7 per cent. For the 1-to-50 emulsion in the deep thin cut, the increase is from 126 to 135 fpm, or 7.2 per cent, and for the 0.050-in. cut at 0.0255 in. feed the increase is from 97 to 108 fpm, or 11.3 per cent. Both sets of curves show that the increase in cutting speed caused by the best cutting fluid over dry cutting is less on the deep thin cuts than for the shallow thick cuts.

It was observed that when turning the monel metal dry or with the emulsions, the chips were continuous and comparatively straight ribbons. The chips curled more as the crater in the tool became deeper. The sulphurized mineral oil caused the chips to coil closer and break off more frequently. The sulphur, however, stained the metal.

Influence of Feed and Depth of Cut

A study was made to determine the influence of varying the depth, feed, or both when turning monel metal using the sulphurized mineral oil (*S.M.* 10). The same standard high-speed

tool of 8-22-6-6-6-15-$^3/_{64}R$ shape was used. The cutting-speed tool-life curves are shown for each of several depths of cut from 0.025 to 0.150 in. for a constant feed of 0.0127 in. per revolution, in the upper half of Fig. 5. This shows the greatest values of tool life for the small depth of cut and the lowest values for the greatest depth of cut. The equations for the different curves are shown. The exponent of T, tool life, is shown to remain constant at 0.105 in all equations for this light feed. It was found that the relation between cutting speed V for a given tool life and the depth of cut d for these lines of constant slope may be represented by a straight line on log-log paper, the equation of which is $Vd^{0.195} = K$, where K is a constant and equal to 78.6 when $T = 60$ min. and $f = 0.0127$ in.

The cutting-speed tool-life curves for a heavier feed of 0.025 in. for each of several depths of cut are shown in the lower half of Fig. 5. The equation for each curve is shown. The exponent of T for the two curves for the 0.025-in. and 0.050-in. cut is the same, indicating the two lines are parallel. The exponent of T in the curve for the 0.100-in. cut is higher, indicating that the curve has a greater slope than the other two. It is shown that for a given value of tool life when the feed is 0.0127 in., by increasing the depth of cut from 0.025 in. to 0.150 in., or 500 per cent, the cutting speed is reduced only 28.8 per cent. If the feed is increased from 0.0127 to 0.0255 in., or 100 per cent, the depth remaining constant at 0.025 in., the cutting speed is reduced only 22.4 per cent. Similarly, for a constant feed of 0.0127 in., a 100 per cent increase in the depth of cut from 0.025 to 0.050 in. results in reducing the cutting speed 14.1

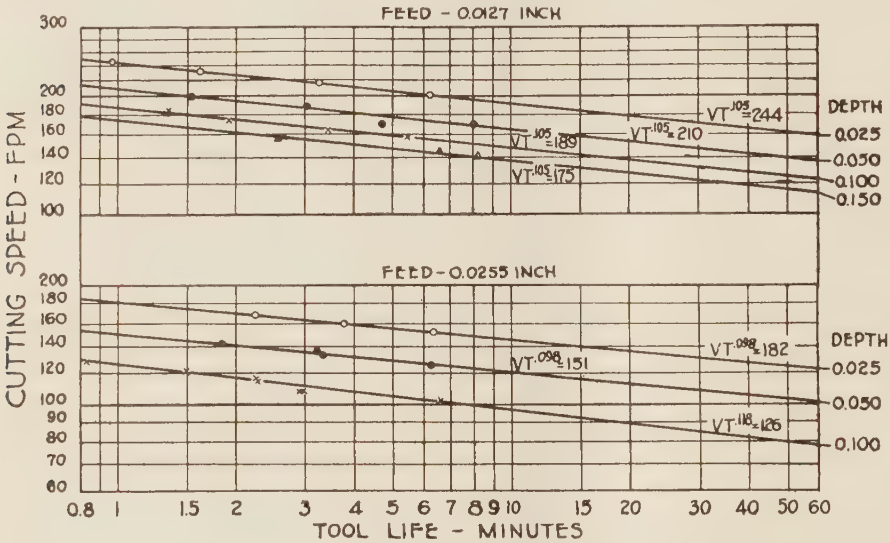

Fig. 5 Cutting-Speed Tool-Life Curves Showing the Influence of Various Feeds and Depths of Cut When Turning Monel Metal

(These curves were obtained when using high-speed-steel tools 8-22-6-6-6-15-$^3/_{64}R$ and a cutting fluid of sulphurized mineral oil.)

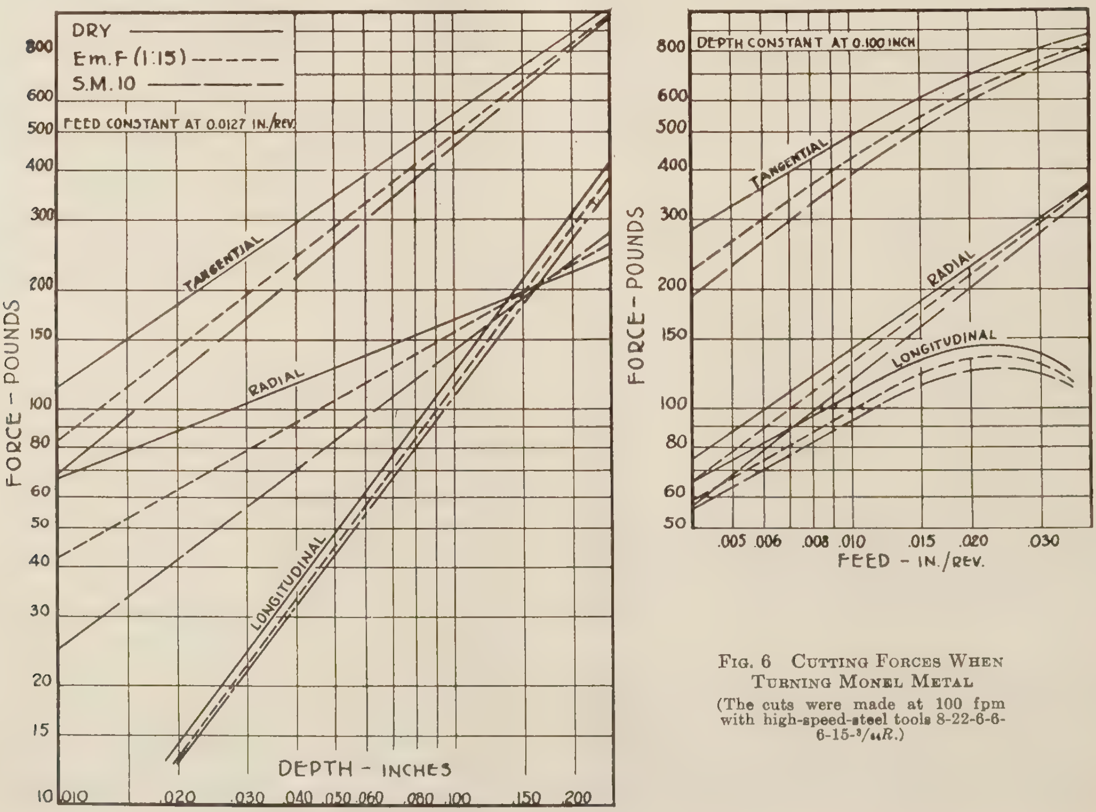

Fig. 6 Cutting Forces When Turning Monel Metal

(The cuts were made at 100 fpm with high-speed-steel tools 8-22-6-6-6-15-$^3/_{64}R$.)

per cent. This again shows, as illustrated in Fig. 3, that the deeper cuts at light feeds are more economical from a tool-life point of view than the shallow cuts with large feeds.

Cutting Forces

Using an all-metal three-component dynamometer, which was described in another paper,[4] the tangential, radial, and longitudinal components of the cutting force were recorded when turning monel metal at different depths of cut and feed. Forces were determined when cutting the metal dry, and when cutting it with the 1-to-15 soap-base emulsion of soluble oil F and with the sulphurized mineral oil *S.M.* 10. The standard high-speed-steel tool ground to the shape of 8-22-6-6-6-15-$^3/_{64}R$ was operated at a speed of 100 fpm. With the feed constant at 0.0127 in. per revolution, readings were taken at various depths of cut ranging from 0.025 in. to 0.150 in. The tangential, radial, and longitudinal forces for each of the three cutting fluids are shown at the left in Fig. 6. All components are highest when cutting dry, they are intermediate when cutting with the emulsion, and are lowest when cutting with the sulphurized mineral oil. When cutting steel it has been found that with everything constant but the cutting fluid the lines representing the tangential, radial, and longitudinal forces are parallel for various cutting fluids. This, however, is not true when turning monel metal. This would indicate that the exponent of the depth d in the equation for the vertical component would be different for each cutting fluid. This equation is usually represented as $F = Cf^x d^y$.

When using a constant depth of 0.1 in. and varying the feed, the force values, as represented by the curves on the right in Fig. 6, were obtained. The curves representing the tangential component of the cutting force are displaced vertically for the different cutting fluids, are not parallel, and are not even straight lines as has been found customary for other metals. It was found that the longitudinal component particularly increased to a maximum as the feed was increased to approximately 0.020 in. per revolution, after which it fell off as the feed was increased to 0.036 in. per revolution. It was found that the cutting forces were least for a newly ground tool. After cutting for a minute or so, the force reached a maximum value where it remained constant over a considerable portion of the life of the tool. The forces shown graphically in Fig. 6 represent these average maximum values. It is felt that the curvature in these various curves shown is due to the cold-working effect of the monel metal, although the phenomena of cold working and the form of the built-up edge were not studied.

The tangential force on this tool turning an S.A.E. 3135 annealed steel dry, as described in another paper,[4] was 340 lb for a cut 0.100 in. deep at a feed of 0.0127 in. For monel metal this force was 550 lb, showing an increase of 61.5 per cent. When the cut was 0.150 in. deep at a feed of 0.0127 in., the forces were 500 and 730 lb, respectively, for the steel and monel, showing an increase of 46 per cent.

Conclusions

As a result of the experiments presented in this report the following conclusions may be drawn for the machining of forged monel metal when being turned with high-speed-steel tools with various cutting fluids.

1 All tools, whether cutting dry, with emulsion, or with a sulphurized mineral oil, seem to fail by cupping on the face and abrasion on the flank below the cutting edge.

2 Tool B shown in Fig. 2, which has a $^1/_8$-inch nose radius and a 15-deg side rake angle, proved superior in dry turning to tool A, which has a smaller nose radius of $^3/_{64}$-in. and a 22-deg side rake angle, for all values of tool life less than 120 min. For shallow thick cuts of 0.050 in. at a feed of 0.0255 in., the straight-edged tool A is superior for a tool life of more than 120 min.

3 A deep thin chip of a given area allowed the higher cutting speed for a given tool life for both tool shapes. When cutting dry, a deep thin 0.100 $\times$ 0.0127-in. cut gave a cutting speed 33 per cent higher than the shallow thick 0.05 $\times$ 0.0225-in. cut, which has the same area as the former. With an emulsion and with a sulphurized oil, speeds approximately 28 and 25 per cent higher, respectively, were obtained for the same tool life when using the deep thin cut as compared to those for the shallow thick cuts under the same conditions.

4 The relation between cutting speed V for any definite tool life and the depth of cut d when cutting with sulphurized mineral oil may be represented, as shown in Fig. 5, by the formula $Vd^{0.195} = K$, where K is a constant and equal to 78.6 when $T = 60$ min and $f = 0.0127$ in. This again shows that it is more desirable in so far as tool life is concerned to use a deep thin cut. By increasing the depth of cut from 0.025 in. to 0.050 in., that is, 100 per cent, the cutting speed is reduced only 14 per cent. By increasing the feed from 0.0127 in. to 0.0255 in., also 100 per cent, the cutting speed is reduced 22.4 per cent.

5 In most cases a rich emulsion of 1 part of soluble oil to 15 parts of water produced slightly longer tool life than the thin emulsion of 1 part of oil to 50 parts of water. This is shown in Fig. 3.

6 It appears that an emulsion is entirely satisfactory as a cutting fluid for turning monel metal. The chips were broken up better when the sulphurized mineral oil was used, but the latter oil stained the metal and it is also more expensive and more unpleasant to work with. When dry cutting, the chips were broken up better by the large-radius tool which, were it not for the fact that the chips rubbed against and scratched the machined finish, would be superior to the straight-edged tool.

7 The machined finish did not seem to be influenced by the type of cutting fluid used.

8 The cutting forces on the straight-edged tool, shown as A in Fig. 2, are as shown in Fig. 6, highest when cutting dry, intermediate when using an emulsion, and lowest when using a sulphurized mineral oil. However, each cutting fluid produced a different value of slope to the tangential-force curves of variable feed or variable depth.

9 It is believed that the tangential cutting force of tool A will not differ appreciably from that for tool B.

10 The tangential force on the straight-edged tool is from 46 to 62 per cent higher when dry cutting monel metal than when dry cutting S.A.E. 3135 annealed steel.

Acknowledgments

Acknowledgment is made to Karl B. Kaiser who assisted in carrying out the investigation. The authors wish to express their appreciation to the International Nickel Company who furnished the monel metal used in the tests. The University of Michigan furnished all equipment and supplies and The Engineering Foundation provided funds with which assistants were employed to run the test and compile the data.

[4] "A Study of the Turning of Steel Employing a New-Type Three-Component Dynamometer," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 58, January, 1936, paper RP-58-1, p. 47.

Comparative Torque and Horsepower Requirements of Standard Four-Flute and Spiral-Flute Taps

By HARRY L. DAASCH,[1] AMES, IOWA

This paper reports certain tapping tests performed as a part of a research project of the Iowa Engineering Experiment Station at Iowa State College.

Data are presented on the torque and horsepower requirements for the use of ½-in., 13-thread, standard-flute, and spiral-tip flute taps during the tapping of S.A.E. 1020 steel. The torque values are shown to vary with speed when a constant percentage of thread is produced. The torque increased with speed when lard oil was used as a cutting fluid; the tendency was toward decreased torque with increased speed when sulphurized oil was used. Spiral-tip taps required less torque than straight-flute taps at similar cutting conditions. Horsepower requirements were found to be less with sulphurized oil than with lard oil at high percentages of thread depths for both kinds of taps. Spiral-tip taps could be operated at lower horsepower inputs than could the straight-flute style when similar cutting conditions exist.

Another series of tests pertains to the dry tapping of cast iron with ⅜-in., 16-thread taps. There was little difference in the torque requirements when the two types of taps were used. Slightly lower horsepower input is shown for the spiral-tip tap.

THE STUDY of metal-cutting dates as far back as the beginning of this century with the work of F. W. Taylor. In recent years a concerted effort has been made in the scientific development of knowledge in the field. It is interesting to note that the title of Taylor's report[2] was "The Art of Cutting Metals." This in itself is suggestive of the fact that any transition from "art" to "science" in metal cutting has been within the past three decades.

[1] Associate Professor of Mechanical Engineering, Iowa State College. Mem. A.S.M.E. Professor Daasch received the degrees of B.S., M.E., and M.S. from Iowa State College in 1925, 1930, and 1933, respectively. He also holds the degree of E.Met. from the Colorado School of Mines, received in 1927. Upon his graduation he was first associated with the Dodge Manufacturing Corporation, later becoming mechanical designer with the Dorr Company, Denver, Colo. In 1927 he became a teaching fellow at the Colorado School of Mines and the following year instructor at Purdue University. After serving as metallurgist with the Hart-Parr Company in Charles City, Ia., Professor Daasch joined the faculty of Iowa State College as assistant professor of mechanical engineering. For the past six years he also served as consultant engineer to various firms primarily in connection with problems of design and production.

[2] "The Art of Cutting Metals," F. W. Taylor, Trans. A.S.M.E., vol. 28, 1907, p. 31.

Contributed by the Machine Shop Practice Division and Special Research Committee on the Cutting of Metals for presentation at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date.

Investigations at Iowa State College have been concerned with the production of threads, and a recent publication[3] has reported torque requirements in the tapping of a low-carbon steel, a gray cast iron, and a cold-rolled brass. The maximum tapping torque when standard taps are used was shown to be dependent upon the combination of metal cut, percentage of thread depth produced, tap speed, and cutting fluid.

The present paper is a report of a series of tests in tapping of gray cast iron and steel with standard straight-flute and special spiral-tip-flute taps. Tapping torque and horsepower requirements are shown for various thread percentages, tap speeds, and cutting fluids.

Materials and Equipment

Taps. Commercial taps were used in these tests. Two styles were used, the regular four-flute straight types and a modified design in which the ends of the flutes were spiral instead of straight. The spiral-flute section is intended to force the chips ahead of the tap. Several modifications of this type are offered by tap manufacturers. All taps had ground threads, were made of high-speed steel, and were secured from open stock without special precautions or specifications.

Metals Tapped. An S.A.E. 1020 steel and a gray cast iron were used. Specifications of these metals are given in Table 1. The steel was purchased in cold-rolled bar lengths $^1/_2$ in. thick. The gray cast iron was cast in slabs 14 in. × 14 in. × $1^1/_2$ in. Test bars $^9/_{16}$ in. thick were slit from the slab, thus producing test-bar sections $^9/_{16}$ in. × $1^1/_2$ in. × 14 in. The thickness of steel test bars was thus equal to the nominal tap diameter, and that of the cast-iron bars $1^1/_2$ times the tap diameter. These values were adopted in accordance with usual machine-design practice.

TABLE 1 METAL ANALYSES

	S.A.E. 1020 steel	Gray cast iron
Total carbon, per cent	0.20	2.98
Graphitic carbon, per cent	...	2.50
Combined carbon, per cent	...	0.48
Silicon, per cent	...	1.76
Manganese, per cent	0.40	0.42
Sulphur, per cent	0.025	0.09
Phosphorus, per cent	0.005	0.67
Brinell hardness no.	126	159

TABLE 2 CUTTING-OIL PROPERTIES

	Lard oil	Sulphurized oil
Viscosity at 100 F	207.2	109.4
Viscosity at 130 F	121.0	65.0
Viscosity at 210 F	56.9	42.2
A.P.I. gravity at 60 F	24.2	23.2
Specific gravity at 60 F	0.9088	0.9147
Flash point (Cleveland open cup) F	430	310
Fire point (Cleveland open cup) F	460	345
Saponification number	196	1.3
Iodine number	57	0.0
Sulphur, per cent	0.65	4.49
Free fatty acid, per cent	4.74	0.00
Color	5	Black

[3] Harry L. Daasch and John Hug. Bulletin No. 123. Iowa Engineering Experiment Station, Iowa State College, Ames, Iowa, November. 1935.

Cutting Oils. The two oils used in these tests were an inedible lard oil and a sulphurized mineral oil. The usual chemical and physical specifications are given in Table 2.

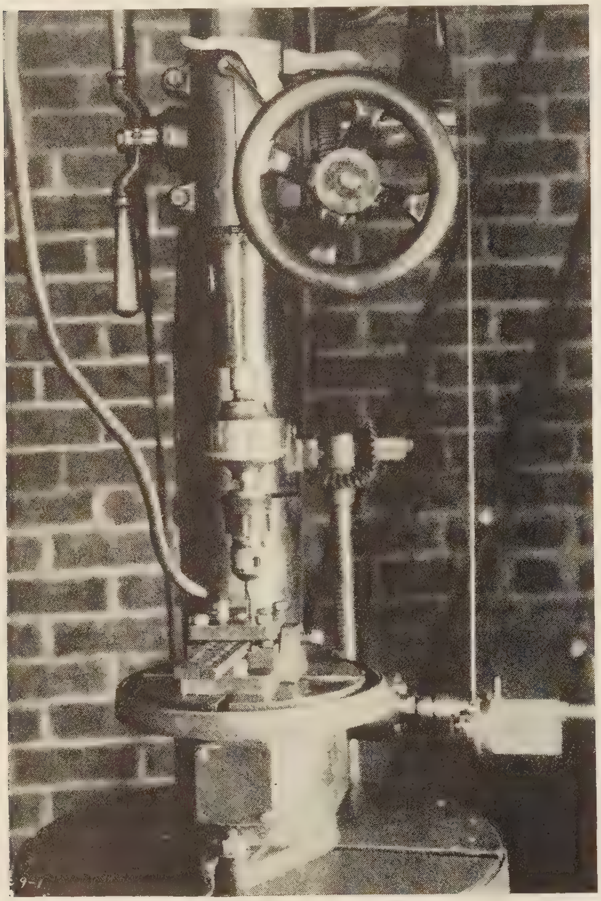

Fig. 1 The Test Apparatus

Drill Press and Tapping Head. A belt-driven 20-in. Excelsior drill press was used in the tests. A range of eight spindle speeds from 25 to 500 rpm was available through back-gear and open belt drives. Power was supplied by a 5-hp motor. A Procunier tapping attachment was used to secure a slight amount of floating of the head and to provide a means of quick withdrawal of the taps from work.

Dynamometer. The dynamometer, which was designed by Prof. John Hug of the mechanical-engineering department at Iowa State College, was set up as shown in Fig. 1. Test bars were clamped to the turntable which was mounted on Radax bearings. A varying force acting on the end of the torque arm prevented rotation of the turntable during the tapping processes. The magnitude of the force was determined by the compression of an ordinary steam-engine indicator spring. A cord from the indicator drum was passed over pulleys to the drill-press-spindle rack. The motion of the indicator stylus in proportion to spring compression combined with the indicator drum turning in synchronism with tap advance or withdrawal resulted in a torque tap-travel indicator card. Typical cards are shown in Fig. 2.

Tests and Calculations

Test bars were drilled to within $^1/_{16}$ in. of required sizes and then accurately redrilled. This procedure was adopted to insure greater accuracy and to eliminate any taper in the holes. Hole sizes were chosen so as to secure a number of increments of thread depths from 30 to 100 per cent. The lard oil and sulphurized oil were used as cutting fluids during steel tapping, and the cast iron was cut dry. Spindle speeds up to 300 and 500 rpm were used in the steel and cast-iron tests respectively.

There was no fixed order or routine in a given series of tests. Random combinations of tap, hole, and tapping speed were considered as a satisfactory precaution against the establishment of trends due to slight variations in metal characteristics, tap-edge conditions, and the like.

Maximum torque values have been determined from measurements of indicator-card maximum ordinates. The conversion of these measurements into torque was accomplished with suitable calibration data.

The indicator-card areas, which were measured with a pla-

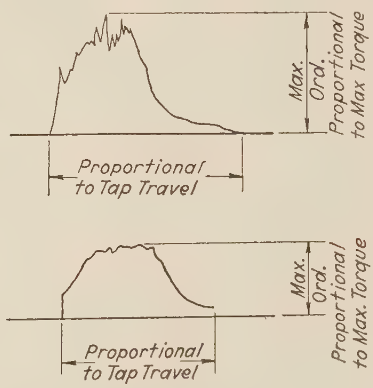

Fig. 2 Typical Cards of Tests
(Above: Tapping of S.A.E. 1020 steel. Below: Tapping of gray cast iron.)

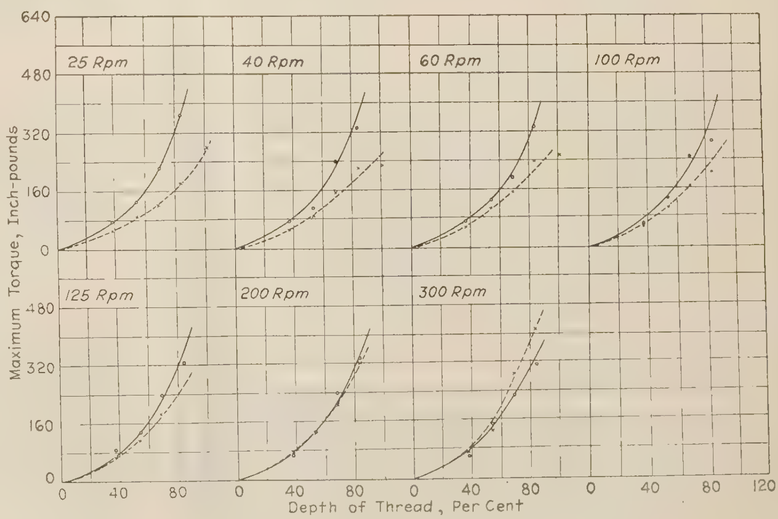

Fig. 3 Relation Between Maximum Torque and Depth of Thread, $^1/_2$-In. 13-Thread, Standard Straight-Flute Tap, S.A.E. 1020 Steel, $^1/_2$ In. Thick
(Dotted lines represent results with lard oil; full lines with sulphurized oil.)

nimeter, were made the basis of power calculations. A standard card length was established on the basis of test-bar thickness plus a length equal to eight times the tap-thread pitch. The period of cutting during the tapping process begins with the entrance of the tap into the hole and continues until the tap projects through the test section for a distance equal to the tap chamfer. Although tap chamfers were equal to six and eight threads and although hole size affects the actual number of tap cutting edges and thus the period of cutting, a standardized length of tap travel was deemed expedient. An average value of four times the tap pitch was therefore adopted for the periods of tap entrance at the top and exit at the bottom. This results in the foregoing bar thickness plus eight times tap-pitch cutting period.

Horsepower values were calculated from the card areas (suitably

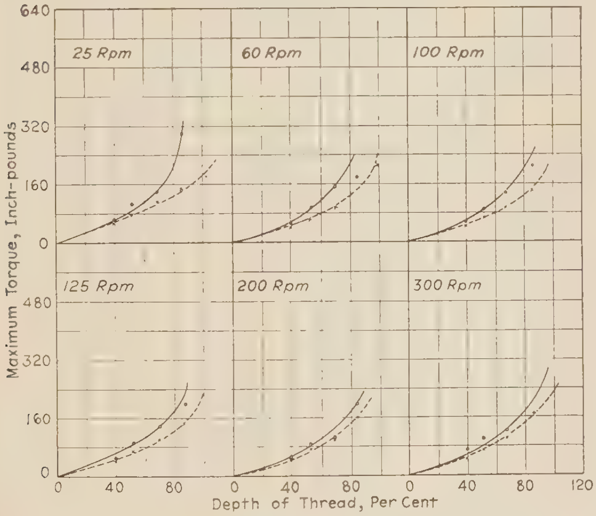

Fig. 4 Relation Between Maximum Torque and Depth of Thread, 1/2-In., 13-Thread, Spiral-Tip-Flute Tap, S.A.E. 1020 Steel, 1/2 In. Thick

(Dotted lines represent results with lard oil; full lines with sulphurized oil.)

converted into energy units) by means of the formula hp $= 2\pi N$ (energy in ft-lb)/33,000, where N is the tap in rpm.

Presentation of Data

Numerous data incident to tests of this kind have been presented in graphical rather than tabular form. Original data pertaining to maximum torque or horsepower were plotted against thread-depth percentage;[4] each plotted point represents the average of as many as six separate sets of data.

The original curves of maximum torque and thread depth and curves of horsepower and thread depth were used as a basis for the construction of a second set of diagrams in which maximum torque or horsepower were plotted against revolutions per minute of the tap.

Information on tapping torque, secured in connection with the use of standard four-flute taps, has been abstracted from Bulletin No. 123.[3] Comparative data for the special spiral-flute taps, as well as all horsepower information for both styles of taps, have been secured by additional series of tests and calculations.

[4] Thread-depth percentages are based on actual tap sizes. A full thread based on nominal tap dimensions is taken as 100 per cent of thread depth. It must be noted that actual tap diameters are slightly greater than nominal dimensions. See Bulletin No. 123 of the Iowa Engineering Experiment Station for discussion of thread-depth calculation methods.

Torque for Tapping of Steel

The maximum torque required during the tapping of S.A.E. 1020 steel with standard and spiral taps is shown in Figs. 3 and 4, respectively. The expected increase of torque with increase in amount of thread cut is shown. It should be noted that at low speeds the torque is higher when sulphurized oil is used than when lard oil is the cutting fluid. This difference becomes less as the tap speed is increased; and in the case of straight-flute

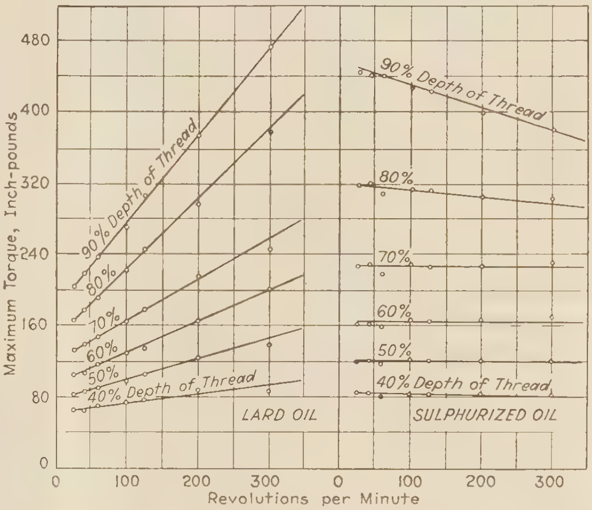

Fig. 5 Relation Between Maximum Torque and Tap Speed, 1/2 In., 13-Thread, Standard Straight-Flute Tap, S.A.E. 1020 Steel, 1/2 In. Thick, Cutting Fluid as Noted

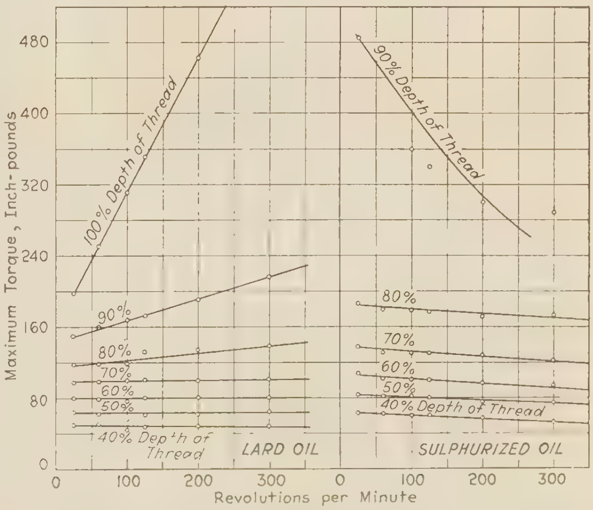

Fig. 6 Relation Between Maximum Torque and Tap Speed, 1/2-In., 13-Thread, Spiral-Tip-Flute Tap, S.A.E. 1020 Steel, Cutting Fluid as Noted

taps the situation is reversed when a speed of 300 rpm is reached at that speed, as shown in Fig. 3, sulphurized-oil cutting conditions require the lesser torque.

Figs. 5 and 6, which have been derived from Figs. 3 and 4, respectively, illustrate the effect of cutting speed on maximum torque. The different torque-speed characteristics, outlined in Bulletin No. 123[3] for straight-flute taps used with lard and sulphurized oils, are shown again when the spiral-tip flute is used.

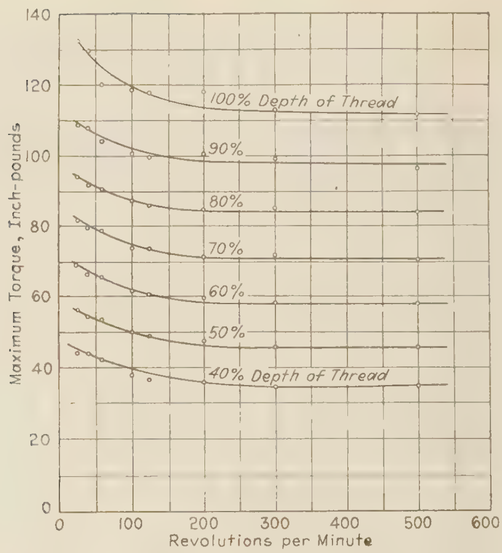

Fig. 7 Relation Between Maximum Torque and Tap Speed, $^3/_8$-In., 16-Thread, Standard Straight-Flute Tap, Cast Iron, $^9/_{16}$ In. Thick, Dry

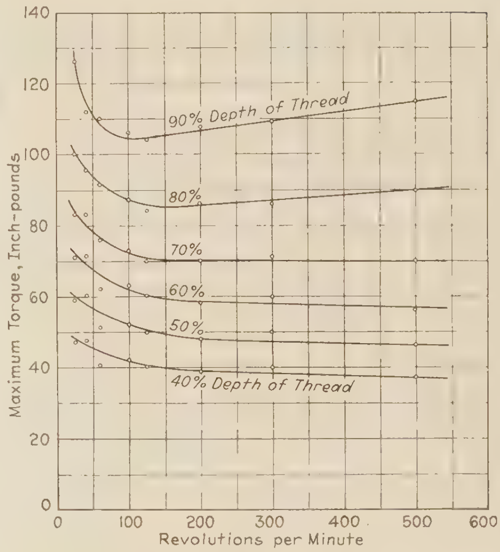

Fig. 8 Relation Between Maximum Torque and Tap Speed, $^3/_8$ In., 16-Thread, Spiral-Tip-Flute Tap, Cast Iron, $^9/_{16}$ In. Thick, Dry

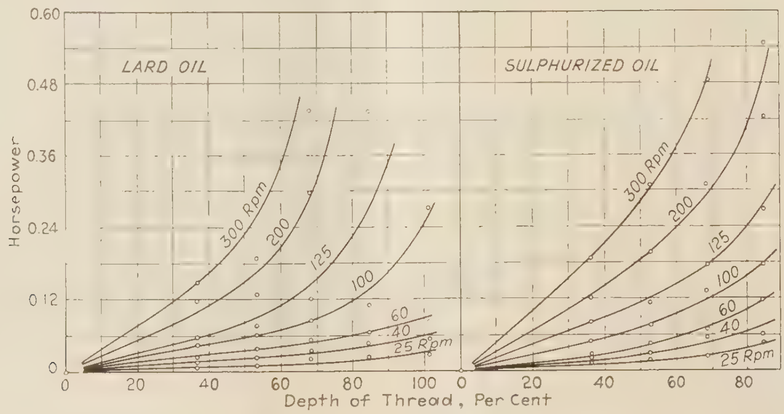

Fig. 9 Relation Between Horsepower and Depth of Thread, $^1/_2$-In., 13-Thread, Standard Straight-Flute Tap, S.A.E. 1020 Steel, $^1/_2$ In. Thick, Cutting Fluid as Noted

Though the purpose of tests outlined here does not include a detailed study of all cutting fluids, the differences of the two oils are of interest. With a straight-flute tap, maximum torque increases with speed when lard oil is used during the tapping of low-carbon steel. This tendency is more pronounced at high percentages of thread. The same trend is shown for spiral-flute taps, but to a lesser degree. Furthermore, the entire range of torque values is lowered when the spiral-flute tap is used. The performance with sulphurized oil is quite different; the tendency, at high percentages of thread production, is toward decreasing torque with increasing tap speed. The general superiority of the special-flute tap is evident.

Torque for Tapping of Cast Iron

Original torque-thread depth curves for the dry tapping of cast iron are not shown here. The derived curves Figs. 7 and 8, show changes of torque with speed. There is slight difference in the results obtained with the two styles of taps. In both cases a small decrease in torque may be expected with an increase of tap speed up to 200 rpm. The torque is constant at tap speeds of 200 rpm or more when straight-flute tools are used. A tendency toward increased torque with increased speed, when high percentages of thread are produced, is shown for the spiral-tip tap.

Horsepower for Tapping of Steel

The horsepower requirements for the tapping of S.A.E. 1020 steel with regular taps and with either lard oil or sulphurized oil are illustrated in Figs. 9 and 10. Fig. 9 shows original data plotted with horsepower and thread depth as coordinates, and these curves have been used in the construction of Fig. 10 where the relation of horsepower and revolutions per minute are shown.

The superiority of lard oil over sulphurized oil is evident at low thread percentages. There seems to be a lesser power requirement with the sulphurized oil at high speeds when more than a 70 per cent thread is cut.

Fig. 11, which applies to spiral-tip taps, may be compared with Fig. 10. A materially less power input is necessary for the spiral-flute tap. The performance of these special taps at low thread percentages is quite the same with both cutting oils. At high percentages of thread depth the horsepower is somewhat less when sulphurized oil is used.

Horsepower for Tapping of Cast Iron

The horsepower required during the tapping of a gray cast iron with the regular and special taps is given in Figs. 12 and 13. In all cases the power varies almost linearly with depth of thread produced and tap speed. The spiral taps require slightly less power for a given condition of thread production.

Discussion

During the process of metal cutting the metal being cut must be separated from the parent metal and the chips must pass or flow over the tool point. The combination of the tearing, deforming, and frictional forces determines the magnitude of tool forces measured by this method of studying machinability. During a continuous cutting operation there will be relatively small variations in the total value of these forces. This applies particularly to turning or planing, and many of the studies reported have been concerned with this type of machining. The cutting tools in turning, planing, shaping, or milling are ground to a form quite generally within the control of the operator or shop management.

A more complicated situation may arise in the use of taps. The simple processes of cutting and chip formation have added to them the discharge of the chips through relatively restricted areas.

When cast iron is tapped, the chips formed are small and well broken. The "as cast" structure, with its dispersed low-strength

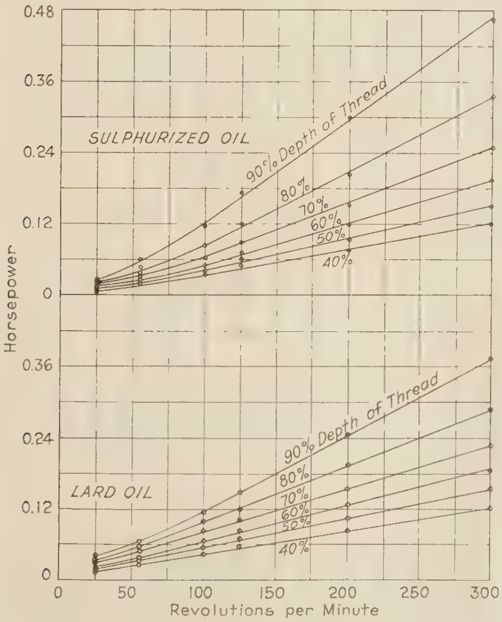

Fig. 10 Relation Between Horsepower and Tap Speed, 1/2-In., 13-Thread, Standard Straight-Flute Tap, S.A.E. 1020 Steel, 1/2 In. Thick, Cutting Fluid as Noted

graphite particles, causes the chip to be formed by a process of compressive failure ahead of the cutting point. The friction due to the passage of the well-broken chip through the tap flute is at a minimum. No exceedingly heavy pressures are involved. The graphite of the cast iron may itself act to reduce friction.

The cutting of the low-carbon steel presents an entirely different problem. Long stringy chips are produced; and during the process of curling out through the tap flute, the chips often "ball" and clog in the flute. The mechanism of cutting is a process of tearing the metal being cut from the parent metal and at some distance ahead of the tool point. Material of higher strength obviously demands greater tool force. Greater friction may be expected as the chip passes over the tool point and curls out through the tap flute.

The test results presented here show the relationship of these facts. The advantage of the spiral flute in the elimination of steel chips is quite evident, as is the fact that chip removal

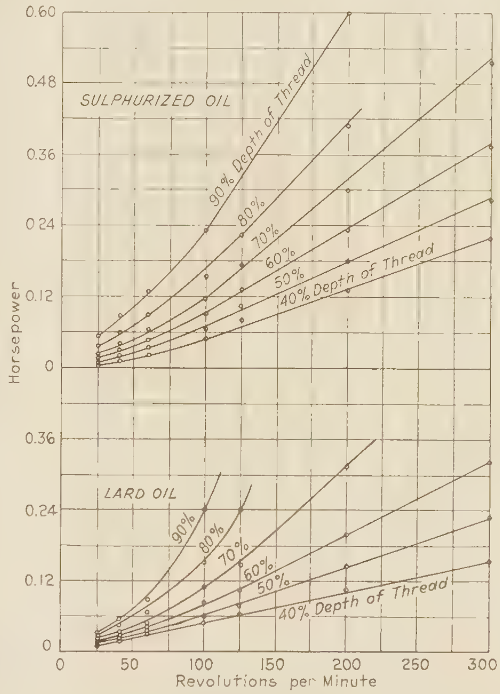

Fig. 11 Relation Between Horsepower and Tap Speed, 1/2-In., 13-Thread, Spiral-Tip-Flute Tap, S.A.E. 1020 Steel, 1/2 In. Thick, Cutting Fluid as Noted

through the flutes is an item of no great importance when cast iron is tapped.

The maximum torque will be somewhat higher than the average torque during the tapping process. Although the energy expended during tapping is proportional to this average torque, it is usually true that an increase in the maximum torque is accompanied by an increase in average torque and power. The relationships between maximum torque, power, and tap life have not been fully investigated. Continuation of the studies along these lines is contemplated.

Conclusions

Specific conclusions which may be drawn from the test results reported are:

1 Maximum tapping torque and horsepower are dependent upon the type of tap flute as well as upon metal cut, tap speed, depth of thread produced, and cutting fluid used.

2 A tap with a special spiral-flute tip which drives the chip ahead of the tap will generally require a lower torque than a standard straight-flute tap when steel is being tapped.

3 An exception to the foregoing is noted when 90 per cent or more thread depth is produced in a low-carbon steel with sulphurized oil as the cutting fluid.

4 An increase in tapping torque with increase in tap speed may be expected with both standard and spiral-tip types of taps when S.A.E. 1020 steel is machined if lard oil is used as the cutting fluid.

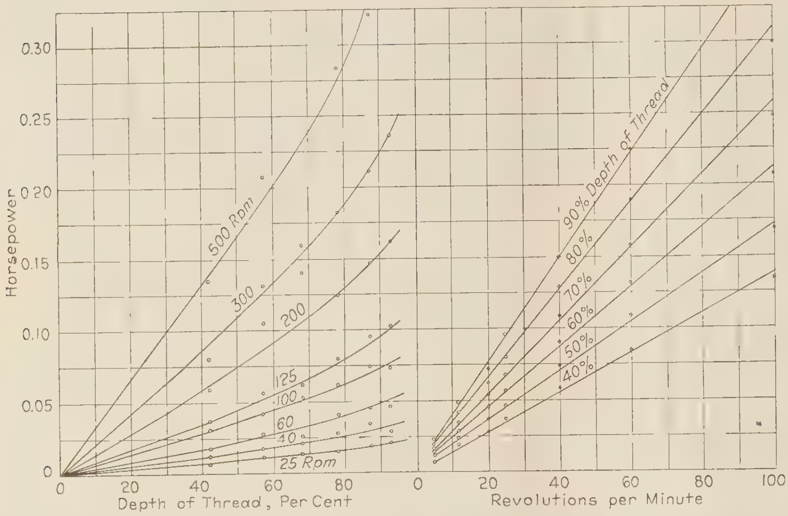

Fig. 12 Relation Between Horsepower, Depth of Thread and Tap Speed, $^3/_8$-In., 16-Thread, Standard Straight-Flute Tap, Cast Iron $^9/_{16}$ In. Thick, Dry

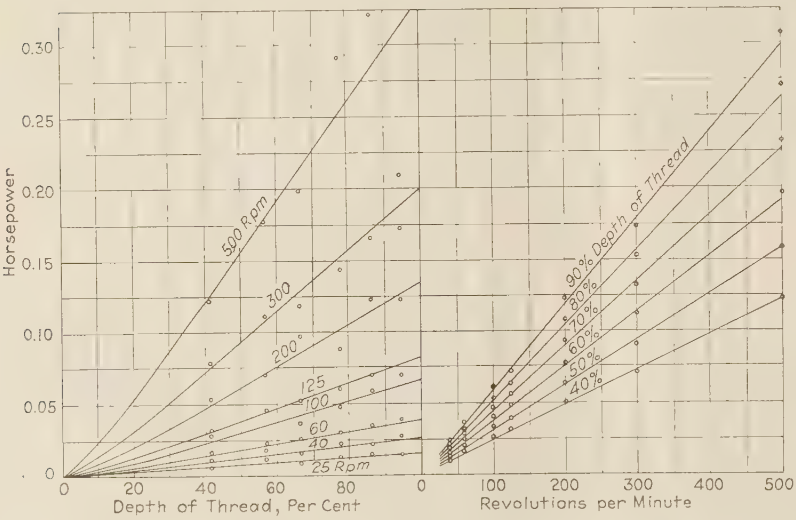

Fig. 13 Relation Between Horsepower, Depth of Thread and Tap Speed, $^3/_8$-In., 16-Thread, Spiral-Tip-Flute Tap, Cast Iron $^9/_{16}$ In. Thick, Dry

5 The foregoing increase is more pronounced in the case of straight-flute taps and when high percentages of thread are produced with either style of tap.

6 A decrease of tapping torque with increase of tap speed may be expected in the production of high percentages of thread in low-carbon steel with sulphurized oil as the cutting fluid with both regular and spiral-tip taps.

7 Spiral-flute taps offer a slight advantage from the standpoint of torque when gray iron is cut.

8 Tapping horsepower increases with increases in thread depth cut and tap speed.

9 A lower horsepower input may be expected in the tapping of high percentages of thread in a low-carbon steel if sulphurized oil rather than lard oil is used as the cutting fluid.

10 The spiral-tip tap requires a lower power than a straight-flute tap when used under similar conditions during the tapping of low-carbon steel and cast iron.

Acknowledgment

The tests reported in this paper were performed as part of a research project of the Iowa Engineering Experiment Station, Dean T. R. Agg, director. This paper is presented with his permission and cooperation. The author acknowledges the cooperative work of Prof. John Hug, who has been active throughout the test work, and to Winter Brothers for their kindness in supplying taps.

Coal Washing and the Baum Jig

By GEORGE L. ARMS,[1] COLUMBUS, OHIO

The author discusses the problems encountered during the preparation of coal by jigs and relates the advances made in this type of apparatus to meet growing demands for increased capacity and greater efficiency. He describes the development of the Baum jig since its introduction in the United States and discusses the operating principles and construction details of the latest design.

THE art of concentration, or separating a granular material into fractions corresponding to a variation in specific gravity, is almost as old as mining itself. A wide variety of apparatus has been employed, ranging from the crude prospector's pan to elaborate and costly processing plants employing tables, jigs, launders, classifiers, and a wealth of accessory apparatus. Processes for each different branch of the mining industry have developed along lines best suited to that particular industry, and while there has been some interchange of practice and apparatus, broadly speaking each branch has developed its own machinery.

In the coal-mining industry, the problem of coal preparation is a difficult one because of the large quantities of material to be handled and the narrow zone in which the specific-gravity separation must be restricted. It is not possible to predicate coal-treatment plants solely on the processing requirements, because of the low unit value of the product after processing. In order to preserve a stable economic setup, the cost of such plants must be kept within limits which will permit amortization. This has resulted in what might be termed "commercial compromises" in the orthodox coal-preparation plant. The industry must develop concentrating apparatus capable of treating the comparatively larger quantities of material peculiar to the coal industry, at higher efficiencies than before, and at the same time keep costs of completed plants within economic limits.

Some measure of the problem confronting the builders of coal-washing plants may be found in a survey of the specific-gravity distribution of the material which constitutes the jig feed. For most practical purposes, the portion of the feed which will float on a liquid with a specific gravity of 1.35 may be considered as pure coal. Likewise, that portion which will sink in a liquid with a specific gravity of 1.60 may be considered as pure waste, since it usually has no fuel value whatever. In the narrow zone between 1.35 and 1.60 specific gravity there are always varying amounts of a middlings material, and it is within this zone that the separation must be made. Specifications for the finished product are being drawn more closely all the time, and it is not uncommon for the builder of a coal-washing plant to be asked to guarantee that the proposed plant will produce washed coal containing less than 1 per cent sink at the washing gravity, and at the same time produce a waste product containing not more than 3 or 4 per cent of float at the same gravity. The exact point within the middlings zone at which separation is to be made is dictated by a study of the coal itself and the market to which it is to be applied, and the relative difficulty of the washing problem is a function of the amount of material which lies at or very near to, the indicated washing gravity.

The demand for this kind of performance, coupled with the necessity for handling capacities running into hundreds of tons per hour, and the economic bar against any extensive reprocessing, has resulted in the design of a multitude of types of coal-washing apparatus.

Perhaps the earliest type of cleaning plant employed what is known as the trough washer, the forerunner of the present day launder or current washer. Other types followed at intervals, and gradually there evolved four major types of apparatus employing water for the cleaning of coal. These are: (1) The launder or current washer, (2) concentrating tables, (3) upward-current classifiers, and (4) jigs. The first three types have had wide application and have been the basis for extensive development work. Many excellent plants both in the United States and abroad make use of one or more of them.

The jigs are numerous and varied. The type may be characterized as a coal washer in which the washing is done by pulsating flow of water upward through a bed of coal, these pulsations alternately lifting and opening the bed, so that settlement can take place, and then closing the bed down on the screen in readiness for another pulsion stroke. The question naturally arises, why pulsate the flow of water? Would not higher capacities be produced if the upward flow were maintained constantly? The answer to that question carries the reason for the popularity and effectiveness of jigs as a general type of coal-washing apparatus. The jigging stroke makes use of the acceleration period in the settling of the particles composing the bed, and it is during this acceleration period that the separation of the particles in accordance with their relative specific gravity takes place.

The earliest jigs were of the movable screen or basket type, where a container with a perforated bottom was jigged up and down in a body of water. The movement of the coal bed relative to the water produced the equivalent of a pulsating flow, and caused stratification. Some of these jigs were equipped with a valve arrangement that caused the basket to act as a pump, thus eliminating the necessity for a separate water-circulating unit.

Then followed the fixed-screen jigs, in which the screen supporting the bed is fixed in the tank and the pulsations of the water produced by means of plungers. These jigs were, generally speaking, more effective than their predecessors, because of the more positive control of the pulsations.

Numerous plants were built employing these types of equipment, and many developments of each type were brought out from time to time. Out of the experience gained in the building and operating of them certain things began to manifest themselves which would need improvement if the jigging of coal were to keep

[1] Construction Engineer, Jeffrey Manufacturing Company. Mr. Arms studied mechanical engineering at Ohio State University, leaving college in his junior year to enter the employ of the Jeffrey Manufacturing Company. Two years later he became connected with the Fairmont Mining Machinery Company where he was employed for two and a half years. He was then associated with the Pittsburgh Coal Washer Company for fourteen years, the Roberts and Schaefer Company of Chicago for seven years, and the Northern Illinois Coal Corporation and the Sunlight Coal Company for three years. He then entered the employ of the Jeffrey Manufacturing Company where he is engaged as construction engineer in connection with coal-washing plants.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

pace with the more stringent operating specifications. In the basket jigs, adjustments were few and hard to make. In the plunger jigs, the same was true to some degree, and as the plants grew older it became more and more apparent that leakage past worn plungers was a bar to consistent operation. With the revival of interest in coal washing during the last few years there has also been a demand for a coal-washing jig with greater flexibility, lower maintenance, higher capacity, and more consistent performance.

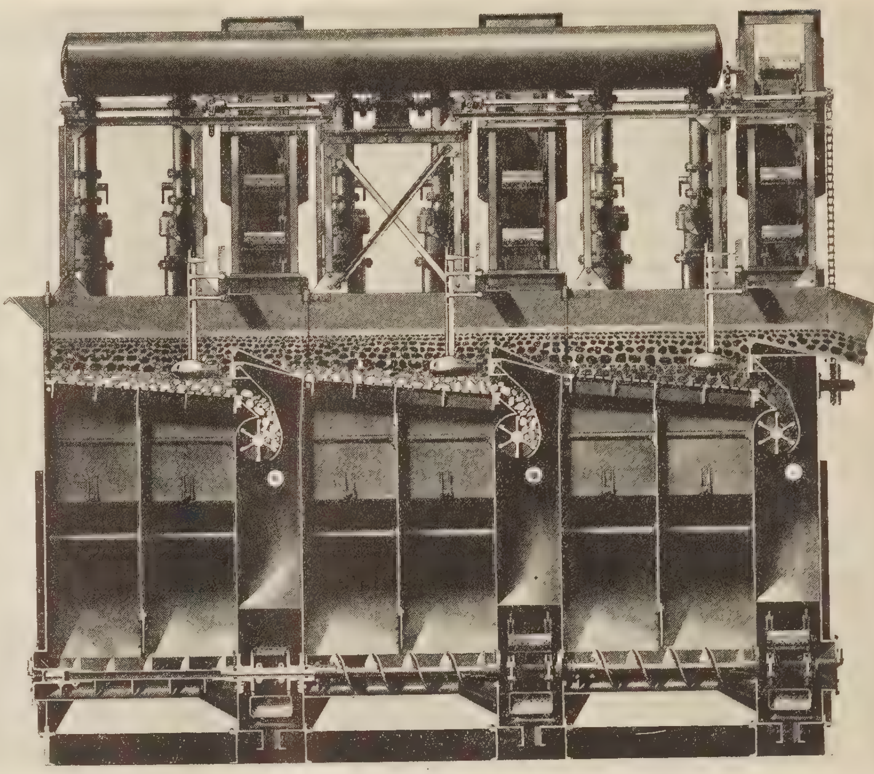

Fig. 1 The Baum-Type Jig for the Preparation of Coal

The jig described in this paper was originated by Fritz Baum in 1892. It is of the fixed-screen type and the characteristic that makes it different from all others of the same type is the use of compressed air instead of a mechanical plunger to produce pulsations. The best analogy for the water action of the Baum jig is a U tube. When air pressure is applied to the surface of the water in one leg of the tube, the water level rises in the other leg, and when the air pressure is released, the water tends to resume a common level under the laws governing a pendulum. If the screen supporting the coal be placed in the leg of the tube opposite where the air pressure is applied, the result will be the equivalent of the Baum jig.

It will be apparent that the characteristics of a stroke produced in this manner will differ from those resulting from a positive mechanical plunger. Investigation of the dynamics involved shows that with no coal in the jig, the action of the water at the jig screen on the downward or suction stroke is analogous to one end of a pendulum swing, the remainder of the cycle being supplied by the energy of the compressed air. It is possible, then, by variation of the volume and pressure of the air to exercise some measure of control over the pulsion or upward stroke through the jig screen. Likewise, by a regulation of the admission of circulating water, in combination with the timing of the air exhaust, it is possible to control the downward or suction stroke through the screen.

The effect of this downward or suction stroke was for many years a debatable subject. It was called "back suction" and was believed by some engineers to be incompatible with the fundamental laws underlying jig operation. In the light of the fuller knowledge and experience of the present day, however, it is apparent that the presence or absence of back suction is of less importance than the ability to control it. The amount of back suction has an important bearing on the ability of the jig to clean fine coal, and there is no question but that the character of the back suction and its controllability in the Baum jig are largely responsible for the greater size ratio in the feed which this unit is capable of handling effectively.

Since the Baum jig was invented and developed in Europe, there were many people who believed that it might not be applicable to the American problem of coal washing. Also a school of thought was developing which emphasized sizing before cleaning as a necessary adjunct to thorough preparation. The Baum jig with its comparatively high cost per machine was not adapted to such a design. The first installations, then, were watched with keen interest, and because of its success the Jeffrey Manufacturing Company, Columbus, Ohio, introduced a redesigned type of the Baum jig in the United States. This design, incorporating operating adjustments not included in the original unit is known as the Jeffrey Baum jig.

Perhaps the most interesting single development was the introduction of a differential stroke to the mechanism producing the pulsations in the flow of water. A study of the behavior of the bed of coal disclosed that jig capacities were limited by the ability of the rising current of water to open up the bed so that settlement could occur. Theoretically, the rising current lifts the bed of coal, at the same time lifting the individual particles away from each other so that settlement can take place according to the laws governing the settling rate of solids in water. Actually, the investigation showed that when a jig began to be pushed for capacity, the particles near the top of the bed were not separating, and that as a consequence, settling was not taking place. This behavior was traced to the fact that with a deep bed of coal, or with a thinner bed and a relatively high velocity of water, there was a tendency to lift the entire bed of coal as a mass, the particles toward the bottom closing together and sealing off the action of the water from those above them. It soon became obvious that if some means were devised to permit infiltration of water into the bed before the actual velocity of pulsation was reached, the bed would be more completely opened, deeper beds could be handled, and higher water velocities would be possible. This gave rise to the differential stroke.

The valve which controls the admission of air to the pressure

compartment in the jig is usually of the piston type, with intake and exhaust ports in the surface of the cylindrical valve body. This form of valve is used in order to obtain the large port areas needed for the passage of comparatively large volumes of air at low pressure. The piston has been split on a horizontal plane, and the two halves have been made independently adjustable. This makes it possible to vary the length of intake time as compared with length of exhaust time, and to utilize the expansion of the air under pressure after the closing of the intake port as part of a completely controlled cycle.

As a check on the effectiveness of this design, a series of tests was run, with varying adjustments of the differential stroke and with varying rates of feed to the test jig. In order to record the effect of the valve adjustment on the flow of water, time-velocity curves were plotted. These curves, in conjunction with a log of the test runs and the analysis of the product, furnish ample data for the study of the effect of this type of stroke. All the tests have borne out the theory underlying its development and although at the present time there are no installations which seriously test the capacity of the new jig, it is believed capable of handling large tonnages with a high efficiency.

In addition to the new design of the air valve, the jig incorporates a provision for readily changing the slope of the screen supporting the coal bed, a hinged plate for keeping the water flow uniform across the width of the jig, adjustments for varying the height of the overflow from each compartment, valves for controlling the amount of air admitted to each cell, and valves for controlling the amount of water flowing to each cell. All these are directly connected, to some degree, with the effect the jig has on the coal passing through it, and making these items adjustable has the effect of increasing the flexibility of the jig as a coal-preparation tool.

Method of Operation

Figs. 1 and 2 show sectional views of a Baum-type jig, and illustrate the method of operation. The air receiver at the top of the transverse section is supplied with air from a centrifugal compressor at a pressure of approximately 2.5 lb per sq in. The pipes leading down from the receiver conduct the air to the valves which admit it to the pressure chamber, the valves being driven by eccentrics mounted on a line shaft directly above the valves. The air pressure forces the water down in the pressure chamber and the shape of the container causes a corresponding rise in the water on the other side of the central partition. This is called the pulsion stroke.

At the end of the pulsion stroke, the air valve opens to the atmosphere allowing the air in the pressure chamber to exhaust. The difference in the water level on the two sides of the partition then causes the water to flow in the reverse direction, and the water rises in the pressure chamber in preparation for another cycle. This is called the suction stroke.

During the pulsion stroke the bed of coal is lifted by the flow of the water and expanded so that the heavier, denser particles can settle to the bottom and form a stratum, or refuse bed.

The fish-shaped apparatus shown in the longitudinal section is one form of float for controlling the discharge of refuse. It is usually streamlined to minimize interference with the movement of coal particles, and its buoyancy is adjusted so that it will sink through the mixture of coal and water but will be supported by

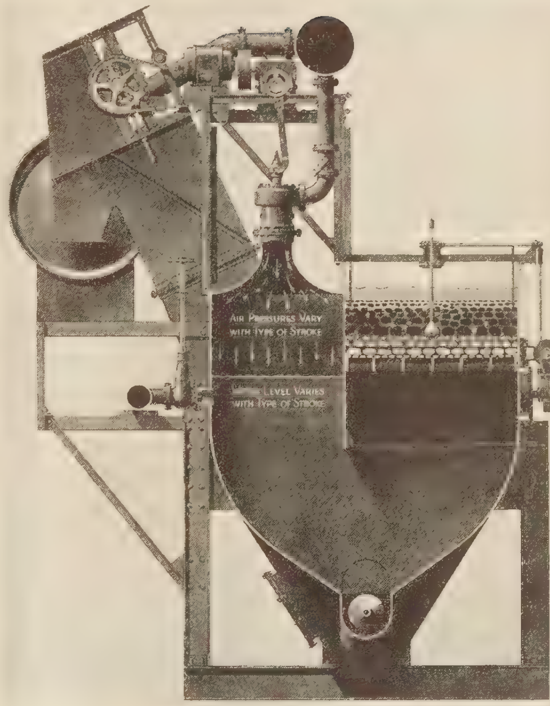

Fig. 2 Side Elevation of the Baum Jig

the denser refuse bed. The float is connected to the control of the refuse-discharge mechanism so that as it rises due to a deeper bed of refuse, it speeds up the rate of withdrawal of this material. Conversely, when the refuse bed thins down, the float slows down the rate of withdrawal, thus preserving the integrity of the bed.

Water is admitted through the pipes and valves shown at the left of the transverse section and flows out over the sluice at the right of the longitudinal section. This horizontal flow of water carries the washed coal out of the jig, as well as assisting in moving the material forward in the jig.

Adjustments to change the operation of the jig are made by varying the flow of water, by varying the air admitted during the pulsion stroke, and by making various changes in certain constructional features such as the slope of the screens supporting the beds, the height of the weirs over which the water flows, and the number of strokes per minute.

Refuse material discharged by the action of the jig settles down through the water into the buckets of the elevators shown in Fig. 1 and is drained by the perforations in the buckets and discharged for further disposal.

The cycle of operations described takes place in each cell of the jig. The complete jig is made up of a number of cells assembled to form compartments, and the compartments grouped to make the complete jig. Thus, in the jig shown in Fig. 1 there are three compartments, each composed of two cells. Various combinations of cells and compartments are used for different applications, this being a function of the amount and quality of the coal to be washed.

Coal Preparation by the Air-Sand Process

By THOMAS FRASER,[1] AURORA, ILL.

The author presents in this paper the features of an air-sand process for the preparation of coal by specific-gravity separation. In this process, the float-and-sink separation is made in an artificial dry liquid medium of sand and air which can be adjusted within certain limits to the specific gravity needed to give a satisfactory coal product. This dry float medium of sand and air is produced by passing a continuous flow of air bubbles upward through a bed of dry sand in a separator box. The author discusses the construction, operation, and effectiveness of the equipment used in the process.

The preparation of coal is the process of manufacturing a commercial fuel to meet market requirements from the raw material obtained in the mining operation. Coal as broken out of the bed and brought to the surface is a natural mixture of good coal particles of all sizes with rock and intermixed particles varying greatly in character and quantity depending upon the nature of the coal bed and associated strata.

In a complete preparation plant, this raw-coal mixture is sized into market grades of uniform-size particles or within definite prescribed size limits, and the refuse particles are removed to produce a finished product as free from impurities and as uniform in quality as the natural characteristics of the raw material will allow. A typical example of the difference in characteristics of coal shipments received without cleaning and shipments from an effective preparation plant is shown by the two ash charts in Fig. 1. The charts show the ash in a long series of individual car samples of coal shipped raw and a series of car samples of mechanically cleaned coal from the same mine. The raw-coal shipments fluctuated in a range of ash content between 5.0 per cent and 9.5 per cent with an average of about 7.0 per cent. The cleaned coal averaged around 4.75 per cent in ash with a much narrower range of fluctuation, between 4.0 and 5.0 per cent. It is manifestly impossible to produce a coal of absolutely uniform quality because it is a natural product with some variation in inherent quality of the clean coal substance and, furthermore, practically all coals as mined contain off-color particles and slightly impure pieces so near to the character of the clean coal as to be incapable of separation from the coal. However, such particles, coated with another coal or flecked with papery films of slate, are more detrimental to the appearance of the product than to its actual heat value. Most commercially workable coal beds in America are capable of producing, by efficient preparation methods, a grade of coal in which a very large proportion of consignments will be within a fraction of 1 per cent of the mean ash content.

Virtually all the commercial processes of coal cleaning make a specific-gravity separation. That is, they take advantage of the fact that the sulphurous and ash-forming mineral constituents of the coal are heavier than the pure-coal matter. Bituminous coal will range from 1.22 to 1.35 in specific gravity; slate from 2.5 to 3.0, and pyrite, the sulphur mineral in the coal, from 4.5 to 5.0. Besides pure-coal particles and pure-refuse particles, all natural raw-coal mixtures contain a third class of material called middlings. These consist of combination particles in varying proportions and ashy bone coals and carbonaceous shales making up a complete series of intermediate specific gravities between the coal at say 1.28 specific gravity and the slate at 2.6 specific gravity. It is this intermediate class of material that constitutes the principal problem in the cleaning of coal and makes it necessary to establish tolerances for the allowable impurities in commercial coal and for fluctuations thereof.

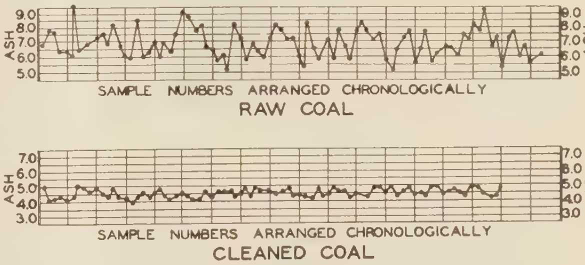

Fig. 1 Ash Fluctuations in Raw Coal and Cleaned Coal

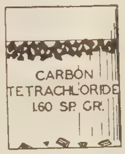

Fig. 2 Principle of Float-and-Sink Test

The standard method of measuring the cleaning characteristics of a coal is the float-and-sink test, by which a sample of the raw material is separated into various specific-gravity fractions for examination. This test method is of interest to be described here because the aim in commercial coal cleaning is to duplicate the effect of these float-and-sink separations at some one specific gravity.

In the float-and-sink test, the sample is immersed in a liquid of high specific gravity capable of floating the coal so that the clean particles collect at the top and the refuse particles sink to the bottom as shown in Fig. 2. These two fractions are then separated by skimming off the float and pouring out the sink. The sample may be separated into a series of specific-gravity classes by using several liquids each of which has a different specific gravity. Thus, by using carbon tetrachloride of 1.60 specific gravity and mixtures of carbon tetrachloride and gasoline to form test liquids of 1.5, 1.4, and 1.3 specific gravity, the sample may be separated into the following classes of particles:

1 The float at 1.3 specific gravity, which comprises the lightest clean coal particles only. These particles will range from less

[1] Mining Engineer, Stephens-Adamson Manufacturing Company. Mr. Fraser was graduated from the University of Illinois, in mining engineering, in 1917, and the following year worked in southern Illinois mines. From 1919 to 1924 he was engaged in coal-preparation developments, as research engineer with the United States Bureau of Mines at Urbana, Ill., and Pittsburgh, Pa. Later he was connected with the University of Illinois as research professor of mining, with West Virginia University as professor of mining engineering, and with the Pennsylvania State Geological Survey in making an estimate and examination of culm and silt accumulations in the anthracite field. Since 1927 Mr. Fraser has been associated with the Hydrotator Company of Cleveland, Ohio, and the Stephens-Adamson Manufacturing Co., Aurora, Ill., where his work has dealt with the development and construction of coal-preparation plants.

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than 1.0 per cent to as much as 6.0 per cent ash in different coals.

2 The 1.3 to 1.4 fraction, consisting of coal particles containing small streaks or slivers of slate or high ashy bone particles. (This class will carry around 9 to 15 per cent ash.)

3 The 1.4 to 1.5 fraction of particles carrying a larger proportion of slate, but still predominantly coal. (This class will carry around 20 per cent ash.)

4 The 1.5 to 1.6 fraction, which in most coals looks decidedly gray in color or in combination particles may be around 25 per cent slate. (This class will carry around 30 per cent ash.)

5 The sink at 1.60, which is heavy rock and pyrites or pieces at least of half rock. (This type of material usually analyzes over 50 per cent ash and often as high as 70 per cent ash.)

Typical float-and-sink data obtained by weighing and analyzing the various specific-gravity fractions of a coal sample are listed in Table 1.

TABLE 1 TYPICAL FLOAT-AND-SINK DATA ON A MIDDLE WEST COAL

Specific gravity	Weight separated, per cent	Ash, per cent	Sulphur, per cent
Float at 1.30	63.0	4.5	0.81
1.30 to 1.40	25.3	8.8	0.84
1.40 to 1.50	4.3	18.2	0.86
1.50 to 1.60	1.8	28.9	0.86
Sink at 1.60	5.6	61.4	2.12

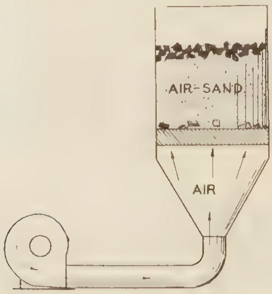

FIG. 3 PRINCIPLE OF AIR-SAND SEPARATOR BOX

In commercial coal preparation, the product is improved by taking out such of the heavier fractions as is necessary and economically feasible to leave a finished product of the desired characteristics. Naturally, as the separation is adjusted with lighter and lighter specific gravities to obtain a better product, the separation becomes technically more difficult and incomplete because we are narrowing the difference in specific gravity between the clean coal and the lightest refuse particles to be taken out. The cost per unit of additional improvement in quality of product increases very rapidly. Thus, when separating with a liquid having a specific gravity of 1.60, a 1.0 per cent reduction in ash can be obtained by removing approximately 2.0 per cent of the product or 40 lb to the ton of 1.60 sink material of 60.0 per cent ash content, whereas to obtain a 1.0 per cent reduction in ash content of the product, by removing a 1.4 to 1.5 class of material of 20.0 per cent ash content, it is necessary to reject about 7 per cent of the product or 140 lb from each ton of raw coal.

In the air-sand process of coal cleaning, the float-and-sink separation is made in an artificial dry liquid medium which may be adjusted, within certain limits, to the specific gravity needed to give a satisfactory product. This dry float medium is a mixture of dry, fine sand and air produced by passing a continuous flow of air bubbles upward through a bed of dry sand in the separator box.

A simple demonstration apparatus, similar to the first device used in development of the process, is shown in Fig. 3. This consists of a cylindrical glass vessel, with a porous-plate bottom and a funnel-shaped air chamber beneath the porous plate, connected to a blower. The porous plate is a uniform air-pervious disk of carborundum or other commercial filter plate, which will diffuse a small quantity of air and deliver it above the plate in a continuous flow of tiny bubbles. These bubbles permeate the sand in the vessel, agitating the entire mass gently and uniformly, to produce a dry quicksand. As long as properly adjusted aeration is maintained, the mass of sand in the vessel is highly fluid and functions as a homogeneous liquid flotation medium. A sample of coal introduced into the vessel separates immediately into a float portion at the top and a sink portion at the bottom, as indicated diagrammatically in Fig. 3. The specific gravity of the fluid is adjusted by varying the rate of air flow.

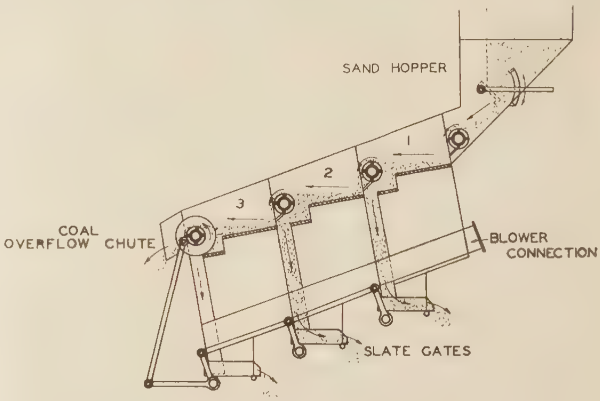

FIG. 4 THE THREE-STAGE AIR-SAND SEPARATOR BOX

To adapt the air-sand fluid to commercial coal cleaning, suitable equipment had to be devised to carry on the float-and-sink operation on a large scale and to discharge the flotant coal product and the refuse sink product continuously. The air-sand separator box, in which this operation is carried out is shown in Fig. 4. This machine really comprises three flotation boxes in series, but all built in one integral trough-like unit separated by intermediate dams with overflow rollers to form the three sand pools designated in Fig. 4 by the numerals 1, 2, and 3. The floor of these sand pools is of air-pervious diffusing plates and the wind box under each of the three sections is separate and supplied with air from manifolds along each side of the machine with individual regulating gates to each of the three compartments of the machine.

In operation, a continuous flow of the aerated sand is maintained in the direction indicated by the arrows. The machine is set up in an inclined position so that the sand fluid runs over it by gravity. Compartment No. 1 is fed continuously at a regulated rate from the sand storage hopper.

This continual flow of sand carries the products through the machine. The flotant coal passes over the rollers and discharges from the machine with the sand, flowing over the roller at the end of the third compartment to the overflow chute. The heavy refuse particles, which sink to the floor of the sand trough, are carried along more slowly by the travel of the sand stream until they reach the vertical sand chutes, one in each compartment, where they pass out of the machine through these discharge chutes to the slate gates below.

These discharge chutes extend transversely across the entire width of the box. The slate gates are short reciprocating feeders that close the bottoms of the refuse chutes and discharge sand at a uniform rate, maintaining a slow downward movement of the sand and refuse columns in the vertical chutes. Feed of

new sand from the storage hopper into compartment No. 1 is regulated to supply sand as fast as it is discharged by the three slate gates and to maintain sufficient overflow at compartment No. 3 to carry the coal away. The reciprocating slate gates are driven by a system of connecting rods and a disk crank on one of the overflow roller shafts.

In the usual method of operation, compartments Nos. 2 and 3 are used as primary cleaner cells while compartment No. 1 is used as a refuse recleaner. That is, the raw coal is fed to compartment No. 2, after which the flotant coal from this pool flows into compartment No. 3 and is again subjected to the flotation effect for an opportunity to sink any refuse particles missed in the first treatment. The overflow from compartment No. 3 is a finished coal product. The refuse from these two primary flotation operations, discharged by slate gates Nos. 2 and 3 is returned with the sand to compartment No. 1 of the separator box and floated again to recover any coal particles that might have been carried down with the refuse in either of the two primary flotation compartments. Only the refuse that sinks in this retreatment operation and passes out through slate gate No. 1 is a final refuse product for disposal to the rock bank.

By this double treatment of both end products, the final separation is made more uniform and foolproof. Temporary maladjustments of intermediate steps in the process are corrected automatically by the retreatment features.

To complete the operation, the sand is screened out of both products and returned to the sand storage hopper, making a continuous closed circuit of the sand medium. The entire equipment is shown in Fig. 5. The coal product is desanded by high-speed vibrating screens which give the coal a substantial agitation at the discharge end to free the coal completely from the sand. The sand which passes through the coal-desanding screens and the refuse-desanding screens goes to the recirculating elevator boot. This elevator also carries the primary refuse returning for re-treatment. Both sand and primary refuse are delivered to the sand hopper and thence to compartment No. 1 of the separator box. The sand is continually conditioned by drying and dedusting in the return circuit.

The most suitable grade of sand to use for the cleaning medium is a washed silica sand of 20 × 100 mesh size. Crushed-quartz glass sand, lake sand, and mixed river sands have all been used successfully, the sole specification being the size. Sand is circulated in the plant at the rate of about $2^1/_2$ tons per ton of coal handled. The make-up sand required to replace losses amounts to between 1 and 2 lb per ton of coal handled.

By adjusting the rate of air flow, the specific gravity of the aerated sand medium may be varied between 1.4 to 1.6. The effective gravity of separation is a little higher than the static specific gravity of the liquid due to the effects of viscosity and the carrying power of the moving currents.

The air supply to maintain aeration of the sand stream is furnished by Connersville blowers, which deliver a constant quantity of air regardless of minor fluctuations of resistance in the sand bed. This enables the operator to maintain a uniform condition of fluidity. The air required varies between 20 and 30 cfm per sq ft of working area, that is, a separator box 12 ft wide with 96 sq ft of porous plates uses about 2000 to 3000 cfm, depending upon the adjustment of the cleaner. A separator of this size will handle coal at the rate of 100 tons per hr. The air pressure in the wind boxes under the air-diffusing plates is 12 to 14 in. of water. The air delivered to the box is controlled by a blowoff valve or preferably by varying the speed of the blower.

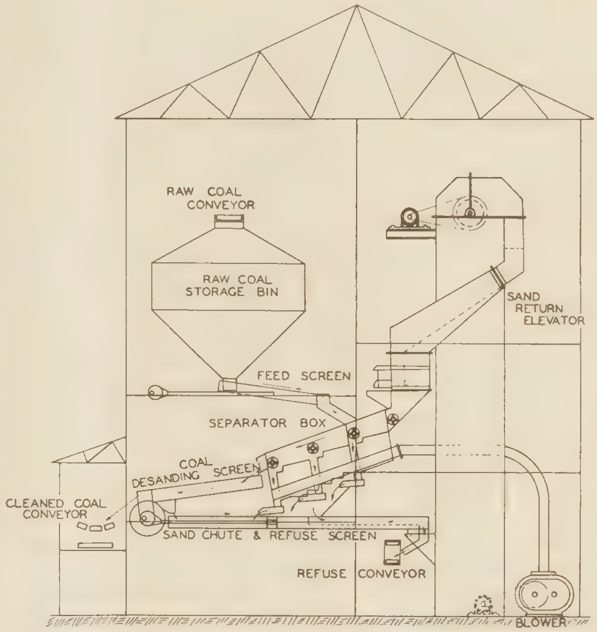

Fig. 5 Typical Air-Sand Coal-Cleaning Plant

The spent air escaping from the separator box is collected by a suction hood over the box. This collection of air, together with excess air from the separator, screens, and other equipment, is sufficient to maintain a satisfactory condition of the plant atmosphere.

This process is adapted to the preparation of egg, nut, and stoker sizes of coal down to $^1/_4$ in. in size. Typical performance of plants in Pennsylvania, Indiana, Kentucky, and West Virginia is shown in Table 2. The cost of cleaning coal by the air-sand process, including operating cost, interest, and depreciation on a 10-year plant-life basis, is around 7 to 8 cents per ton on predominantly dry coal. If predrying of all or part of the coal is necessary, this will add from 1.5 to 5 cents per ton, depending upon the extent of the drying problem.

TABLE 2 TYPICAL PERFORMANCE DATA OF AIR-SAND CLEANING PLANTS

	Raw coal			Cleaned coal			
Coal bed and location	Ash, per cent	Sulphur, per cent	Moisture, per cent	Ash, per cent	Sulphur, per cent	Moisture, per cent	Ash in refuse, per cent
Lower Kittanning, Pa.	12.5	2.4	2.4	7.3	2.00	2.30	61.0
Lower Freeport, Ind.	9.2	...	2.2	7.0	...	2.00	50.5
Green River, Ky.	10.2	3.5	7.0	7.3	2.50	7.00	65.0
Chilton, W. Va.	7.2	0.9	1.5	4.6	0.76	1.25	52.0

The Rheolaveur Coal-Cleaning Process

By JOHN GRIFFEN,[1] PITTSBURGH, PA.

The author discusses the Rheolaveur process of cleaning coal, explains its operating principles, and relates how variations in raw-coal qualities are smoothed out in cleaned-coal product. He submits data showing the variability of raw coal from Illinois and Pittsburgh seams, and the uniformity of the cleaned coal produced in the Rheolaveur plants.

THE CLEANING of raw coal is done not only to improve its average qualities but also to increase the uniformity of the product. Whether the coal is used in metallurgical processes or as steam-plant or domestic fuel, uniform qualities are desirable. In fact, uniform qualities are very often more important than average qualities.

In steam-boiler practice it is much more essential that the coal be uniform in ash and sulphur content, heating value, and ash-fusion temperature than to be low in average ash and sulphur content and high in average heating value and ash-fusion temperature. The fuel-burning equipment and furnace of any boiler can be constructed to burn almost any type of coal efficiently, but the highest boiler rating and efficiency and the lowest maintenance costs can be obtained only when the coal supplied conforms closely at all times to the standard of qualities for which the equipment and furnace were designed.

In metallurgical processes uniformity is, if anything, more important. The reactions occurring in such processes are numerous and complex. The smooth working of a blast furnace is largely dependent upon the uniformity of the materials in the burden. Undue variability in the coke is reflected in uneven operation and variable quality of metal. Seyler[2] says: "The improvement of the physical properties of the metallurgical coke (from washed coal), in addition to the reduction of ash and sulphur, has been reflected in blast-furnace practice to the extent of a 5 to 8 per cent reduction in coke consumption, a 5 to 10 per cent reduction in flux with a subsequent reduction of 7 to 12 per cent in slag volume, a 5 to 8 per cent reduction in blast pressure, resulting in a substantial reduction in power requirements, and a 5 to 8 per cent increase in production of a low-sulphur pig iron. These improvements in blast-furnace practice, under the same operating conditions, were based on the average operation of five different plants which changed from coke made from raw coal to that made from washed coal." It is undoubtedly true that the increased "production of low-sulphur pig iron" was caused by greater uniformity in coke qualities as well as by the improvement in average coke quality.

One of the features of the Rheolaveur coal-cleaning process is the continuous recirculation of an intermediate-gravity product, the purpose of which is to produce efficient cleaning results and uniformly cleaned coal. This method of utilizing an intermediate-gravity product gives a regulating effect like that of a flywheel, so that the variations in raw-coal qualities are smoothed out in the cleaned coal. It is the purpose of this paper to describe the Rheolaveur process briefly and submit data showing the variability of raw coal of various types and the uniformity of the cleaned coal produced in Rheolaveur plants. Due to the processes involving the formation of coal seams, the materials therein are quite variable and often intermixed so that the quality of the material may vary from fairly pure coal to refuse containing little or no combustible material; the purer combustible material is called coal, the more impure is called bone or middlings, and material containing little or no combustible is called refuse. Practically all commercial coal-cleaning equipment and processes are based on the fact that coal has a low specific gravity relative to refuse materials and that, generally, as the specific gravity of the individual pieces of material found in the output of a coal mine increase, their combustible or coal content decreases. Hence, by separating and discarding the materials with heavier specific gravities, the coal can be loaded in a relatively clean condition.

Primarily the problem in coal cleaning is the control of the material with intermediate specific gravities. Its efficient separation into that part of better quality which may be shipped with the cleaned coal and that part of poor quality which should be rejected with the refuse, determine the effectiveness of the plant as to overall recovery and the quality and uniformity of the cleaned coal.

The Rheolaveur process utilizes launder washers where the raw coal, flowing down a launder or trough in a stream of water, is stratified in accordance with differences in specific gravity so that the stream consists of the light material flowing rapidly in an upper layer, a middle layer of heavier material moving more slowly, and a bottom layer of the heaviest material flowing quite slowly. This flow of materials is illustrated in Fig. 1. Rheo boxes, which control the extraction of the materials with the heavier specific gravities (refuse and bone) are attached to the bottom of the launder. These boxes are provided with water connections so that an upward current of water of controlled volume will classify the material delivered to the box from the launder. This classification supplements the separation effected by the flow in the launder.

The system of launders and boxes is arranged so that the intermediate gravity or bone material is collected and recirculated within the launder, thus, increasing the thickness of the middle layer stratified in the launder. The middlings return also supplies the fly-wheel action mentioned previously. This action absorbs variations in tonnage and quality of raw coal. With such

[1] American Rheolaveur Company and the Koppers-Rheolaveur Company. Mr. Griffen was graduated from Lehigh University in 1911 as a chemical engineer. The following year he was assistant fuel engineer, Lehigh Coal and Navigation Company, Lansford, Pa., in charge of briquetting plant; from 1912 to 1915 he was assistant to the president, Harrison Bros. & Company, Inc., Philadelphia, Pa., chemical manufacturers, supervising their mining operations on bauxite, barytes, and pyrite. The next three years he spent as fuel engineer, The Hudson Coal Company, Scranton, Pa., in charge of inspection of their shipments of anthracite coal, and engaged on experimental work in the use of pulverized anthracite for stationary and locomotive boiler use. From 1918 to 1926 he was manager, anthracite territory, The Dorr Company, Wilkes Barre, Pa. Since 1926 Mr. Griffen has been associated with the American Rheolaveur Company and the Koppers-Rheolaveur Company engaged in the design, erection, and operation of coal-cleaning plants for both anthracite and bituminous.

[2] "Washing Coal for Coking Purposes at Clairton By-Product Coke Works," by H. G. Seyler, *Coal Age*, vol. 38, June, 1933, p. 187.

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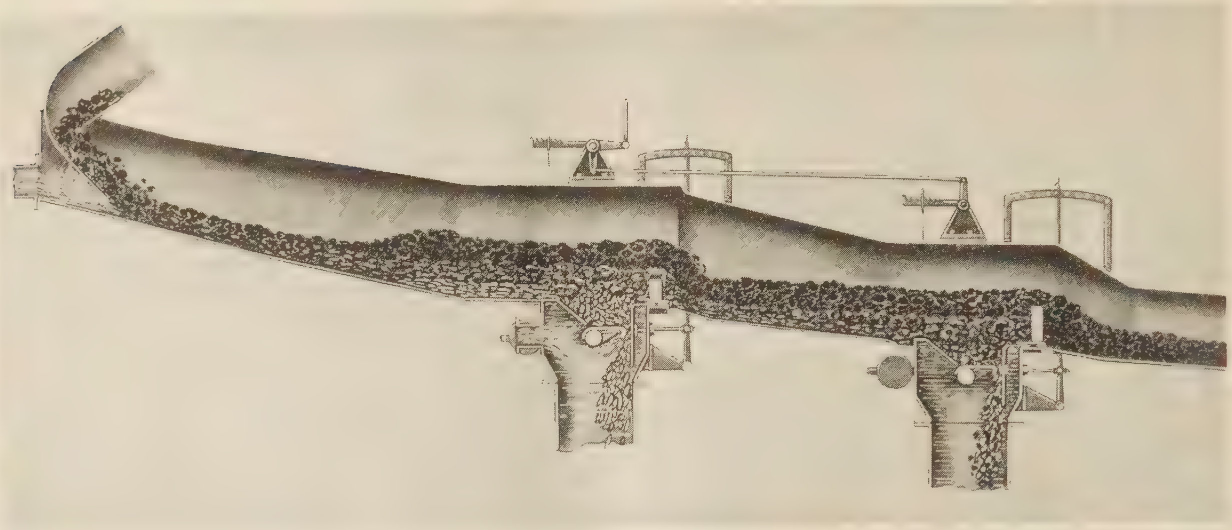

Fig. 1 Stratification of Material in the Rheolaveur Coarse-Coal Cleaning Plant

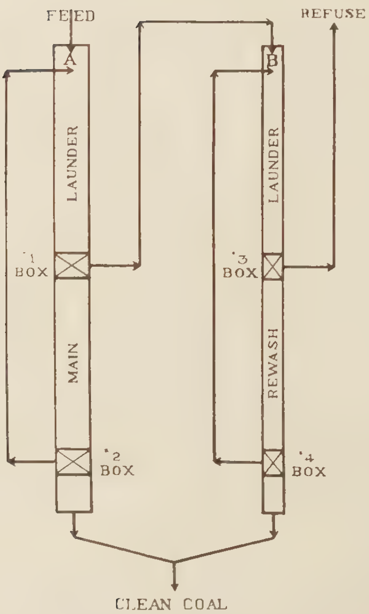

Fig. 2 Plan Flow Diagram of the Rheolaveur Coarse-Coal Cleaning Plant

an arrangement, the separation of refuse is always made between refuse and middlings, and the separation of coal is made between coal and middlings. This system permits the cleaning of unsized coal over a wide range of sizes.

The individual units of Rheolaveur apparatus, that is, the launders and boxes, are assembled in various definite combinations to meet the requirements of different raw coals, of the markets the clean coal is to serve, and of the tonnage and size range involved. Rheolaveur plants may be simple or elaborate as the conditions may require.

With all concentrating and coal-cleaning equipment, an increase in the range of sizes handled in one unit or an increase in the amount of material having a specific gravity approximating that at which the separation is to be made, increase the difficulty of making a sharp separation in all sizes at the specific gravity selected. To meet this condition a wide range of coal sizes is screened into two or more sizes. One type of Rheolaveur unit has been developed for large or coarse coal and another for small or fine coal. Flow diagrams of typical units of each type are shown in Figs. 2, 3, and 4. Fig. 3 is a flow diagram in elevation of the same unit shown in plan in Fig. 2. Fig. 5 shows a section of a fine-coal Rheolaveur launder and box and illustrates the stratification and classification of this coal. These two types of units in combination constitute the complete plants for which performance data are given hereafter.

Uniformity of Coal

The uniformity of coal may be expressed by recording its

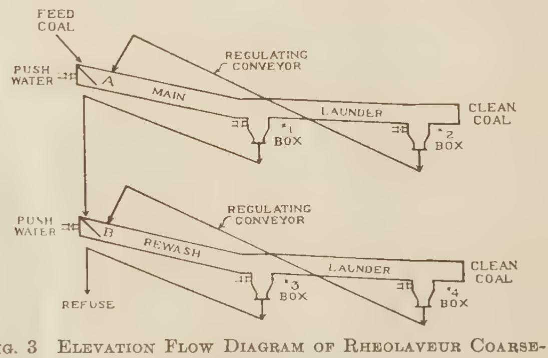

Fig. 3 Elevation Flow Diagram of Rheolaveur Coarse-Coal Cleaning Plant

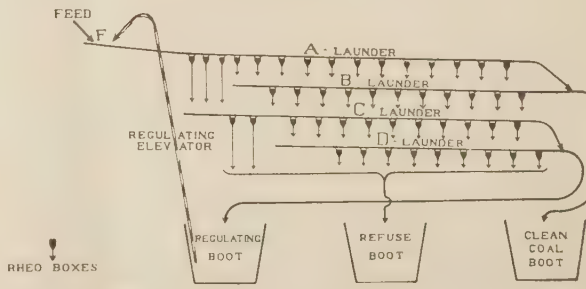

Fig. 4 Elevation Flow Diagram of Rheolaveur Fine-Coal Cleaning Plant

variability. A uniform coal has a low degree of variability. The measures of coal quality generally used in industry are moisture, ash, sulphur, heating value, and the ash-fusion temperature. As ash content is the most simple measure of coal quality it will be used in this paper. A more complete presentation of the variability of coal qualities has been given by Morrow and Proctor.[3] However, a quotation from page 4 of the article[3] will throw light on what is presented hereafter:

"It is generally accepted that the results of a series of samples will illustrate the degree of variation of the coal itself. Rather

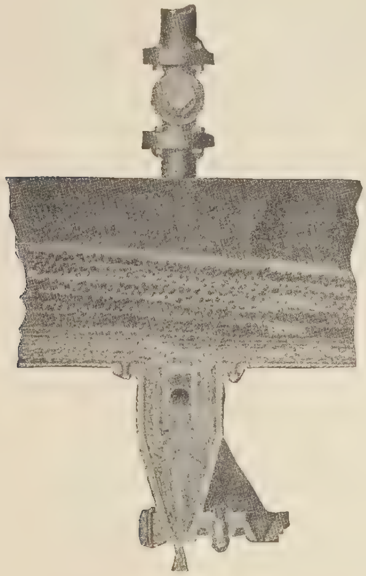

Fig. 5 Fine-Coal Rheo Box Showing Stratification and Classification of the Coal

the results should be considered as seeming variations due in part to the true variations of the coal and in part to the sampling and analysis. These variations depend on the following factors, which must be considered in any study that undertakes to reveal the true quality and variability of a given coal: (1) The true variability of the coal itself, (2) the accuracy of taking the gross sample, (3) the accuracy of reducing the gross sample to minus 4 mesh, (4) the accuracy of reducing the minus-4-mesh sample to minus 60 mesh, and (5) the accuracy of the analysis of the sample.

"In many cases we have found that the last four factors may cause more variation in the results than that caused by the true variability of the coal itself."[3]

In judging the performance of a coal-cleaning plant the variability introduced by the method of sampling and analysis should be minimized if the same procedure is used for raw coal and cleaned coal. This condition has been met so far as the writer could determine in every case where a comparison has been here drawn, unless any variation in sampling method is described.

To express variability, the usual practice is to show the percentage of samples that fall within various limits from the average, expressed graphically or by tabulation. It has been shown by Bailey[4] and the British Engineering Standards Association[5] that the distribution of sampling results follows closely the laws of probability. Morrow and Proctor[3] have confirmed this in their work. Thus, the determination of the probable error of a set of samples fixes the distribution curve of the variations, and it is possible to express the distribution by this one figure. This does not determine any possible skewness in the distribution, but does give a convenient and simple number for comparison. Morrow and Proctor have discussed the skewness tendency of coal analyses and on page 9 of their article[3] state that "the largest variation from the average ash or sulphur should be on the plus side, but on the minus side for ash fusion" (temperature).

By definition, the probable error is the value of the plus-and-minus limit from the average within which 50 per cent of the results will fall, and the probable error r denotes the distribution as given in Table 1.

TABLE 1 VARIABILITY OF COAL DENOTED BY PROBABLE ERROR r

Limits	Percentage of results
Average $\pm r$	50.0
Average $\pm 2r$	82.3
Average $\pm 3r$	95.7
Average $\pm 4r$	99.3
Average $\pm 5.7r$, maximum error	Expected once in 10,000 results

TABLE 2 VARIABILITY OF ASH CONTENT OF RAW AND CLEANED COAL FROM AN ILLINOIS STRIP MINE, PLANT A

Size of coal, in.	1 1/2 to 0		1 1/2 to 1/4		1/4 to 0	
Type of coal	Raw	Cleaned	Raw	Cleaned	Raw	Cleaned
Number of cars from which samples were taken	396	110	417	287	155	108
Ash, dry basis, per cent:						
Average	14.48	5.02	10.94	4.44	21.86	4.05
Maximum	28.45	9.28	27.62	8.96	33.40	5.93
Minimum	6.76	2.47	4.03	2.40	10.78	2.59
Probable error	2.25	0.79	2.14	0.68	2.46	0.51

Uniformity of Rheolaveur Cleaned Coal

The data on which the following portion of this paper is based covers (1) 2600 analyses of raw and cleaned coal from two Rheolaveur plants cleaning two different Illinois coals mined by stripping and (2) a large number of analyses of coal from similar plants cleaning coal mined from the Pittsburgh seam.

Plant *A*—Illinois Strip-Mine Coal

At this property coal is produced primarily for steam-plant and domestic use and is produced from three pits, all mining the same seam. Every car of slack and nut slack produced from this property is separately sampled and analyzed for moisture and ash. Sampling is done by cutting the stream of coal entering the car. Four increments of 20 lb each are taken from a 50-ton car and six such increments from larger cars. These increments are spaced equally during the loading of the car. The same procedure was continued for the cleaning-plant product.

The coal-cleaning plant consists of a coarse-coal unit such as shown in Figs. 2 and 3, which receives 3-in. to 0-in. coal. Its clean-coal product is screened into the nut sizes from 3 to 1/4 in., and the 1/4-in. to 0-in. coal is recleaned in a fine-coal unit similar to that shown in Fig. 4.

The analyses covering the output of raw coal for the three months prior to the starting of the cleaning plant and for the first three to four months of its operation are given in Table 2. The distribution from the average for the raw and cleaned 1 1/2-in. to 0-in. screenings, 1 1/2-in. to 1/4-in. nut, and 1/4-in. to 0-in. duff, are given in Figs. 6, 7, and 8, respectively.

This coal, due to its high ash content and high probable error in the raw state makes a very interesting comparison. The

[3] "Variables in Coal Sampling," by J. B. Morrow and C. P. Proctor, Technical Publication No. 645-F-67, American Institute of Mining and Metallurgical Engineers, September, 1935.

[4] "Accuracy in Sampling Coal," by E. G. Bailey, *Journal of Industrial and Engineering Chemistry*, vol. 1, March, 1909, p. 161.

[5] "Report on the Sampling of Small Fuel up to 3 In. Embodying Some General Principles of Sampling," by E. S. Grunnel and A. C. Dunningham, Report No. 403, British Engineering Standards Association, December, 1930.

tendency to skewness on the high-ash side of the raw coal is very pronounced. The cleaned coal shows much greater uniformity; the probable error being one third to one fifth of that of the corresponding size of raw coal. The skewness of the cleaned coal on the high-ash side is very slight, and with the $^1/_4$-in. to 0-in. size, duff is negligible. It is interesting to note that this smallest size is cleaned to the lowest average ash content and the lowest probable error, i.e., the highest uniformity in the product. Small coal is generally considered more difficult to clean than large coal.

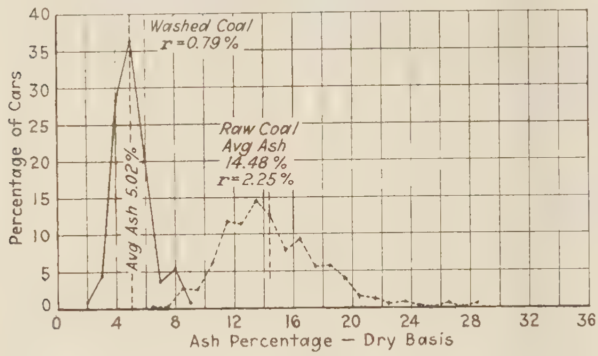

Fig. 6 Distribution From the Average for Raw and Cleaned $1^1/_2$-In. to 0-In. Screenings From an Illinois Strip Mine, Plant *A*

(Values plotted were obtained from analyses of washed coal from 110 cars and raw coal from 396 cars.)

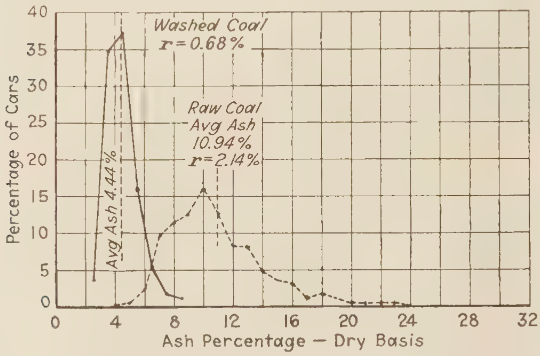

Fig. 7 Distribution From the Average for Raw and Cleaned $1^1/_2$-In. to $^1/_4$-In. Nut Coal From an Illinois Strip Mine, Plant *A*

(Values plotted were obtained from analyses of washed coal from 287 cars and raw coal from 417 cars.)

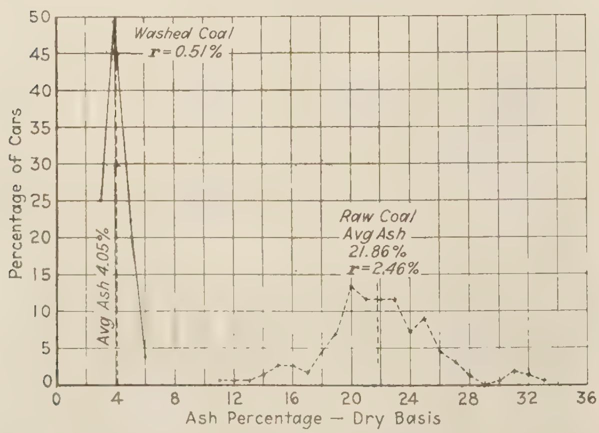

Fig. 8 Distribution From the Average for Raw and Cleaned $^1/_4$-In. to 0-In. Duff Coal From an Illinois Strip Mine, Plant *A*

(Values plotted were obtained from analyses of washed coal from 108 cars and raw coal from 155 cars.)

Plant *B*—Illinois Strip-Mine Coal

This property likewise produces coal for steam-plant and domestic use from three pits mining one seam. No organized and continued analysis program was carried out on the shipments of raw coal, so these data are not available.

The coal-cleaning plant consists of a coarse-coal unit similar to that illustrated in Fig. 9, which receives raw coal of 4-in. to 0-in. size and screens its cleaned coal into nut sizes from 4 in. to $^5/_{16}$ in. The $^5/_{16}$-in. to 0-in. coal is recleaned in a fine-coal unit similar to that in Fig. 4. Since the raw coal larger than $^3/_4$ in. contains considerable laminated intermediate-gravity material which cannot be shipped and contains appreciable coal which is liberated by crushing, the screen and crusher are used to crush the coarser part of the coal product from the rewash launder in which this material is concentrated. The crushed material is returned to the original feed point and recleaned.

Samples are obtained by cutting with a swing box a single increment sample from the stream of coal entering the railroad cars. The samples of $1^1/_4$-in. to 0-in. screenings, weigh approximately 50 lb, samples of $^3/_4$-in. to 0-in. screenings weigh about 35 lb, and samples of $^5/_{16}$-in. to 0-in. screenings weigh about 20 lb. Each sample is analyzed separately for moisture and ash. Samples are taken about every hour depending on the loading schedule of the various sizes. During each 7-hr shift, from three to eight samples of each size are taken and analyzed. The results of all analyses obtained over seven months' operation are recorded in Table 3 and are plotted in Figs. 10, 11, and 12.

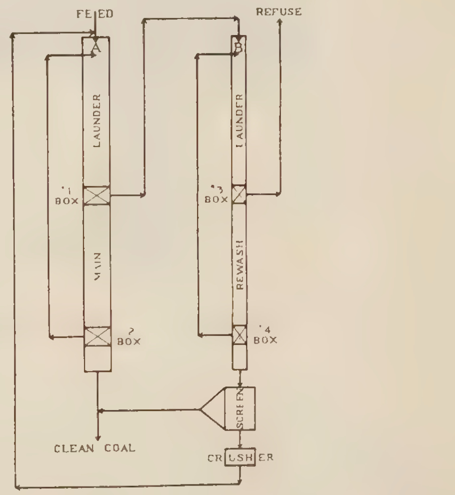

Fig. 9 Plan Flow Diagram of Rheolaveur Coal-Cleaning Plant Used at an Illinois Strip Mine, Plant *B*

It is unfortunate that similar data on the raw coal are not available for comparison. These ash-distribution curves on Rheolaveur cleaned coal show a low variability for the type of

TABLE 3 VARIABILITY OF ASH CONTENT OF CLEANED COAL FROM AN ILLINOIS STRIP MINE, PLANT *B*

Size of coal, in.	$1^1/_4$ to 0	$^3/_4$ to 0	$^5/_{16}$ to 0
Number of samples[a]	489	277	367
Ash, dry basis, per cent:			
Average	9.46	9.40	8.12
Maximum	11.00	11.15	9.65
Minimum	7.95	7.90	6.60
Probable error	0.36	0.35	0.31

[a] Samples taken hourly.

TABLE 4 VARIABILITY OF ASH CONTENT OF RAW AND CLEANED COAL FROM THE PITTSBURGH SEAM, PLANT *C*

Size of coal, in.	2 to 0		2 to $1^1/_8$		$1^1/_8$ to 0		$1^1/_8$ to $^3/_8$		$^3/_8$ to 0		4 to 0	
Type	Raw	Cleaned	Raw	Cleaned	Raw	Cleaned	Raw	Cleaned	Raw	Cleaned	Raw[a]	Cleaned
Number of cars from which samples were taken	..	100	..	100	100	100	..	100	..	100	...	.....
Ash, dry basis, per cent:												
Average	..	7.90	..	7.50	11.90	8.20	..	7.60	..	8.60	12.00	.....
Maximum	..	9.10	..	9.40	18.00	10.10	..	9.20	..	10.00	15.40	.....
Minimum	..	6.50	..	6.10	8.30	7.00	..	6.10	..	7.80	9.10	.....
Probable error	..	0.36	..	0.59	1.36	0.36	..	0.45	..	0.30	1.01	.....

[a] The 4-in. to 0-in. raw coal was not sampled at the cars but at the feed point in the cleaning plant during three operating days when 102 samples of 110 lb each were taken and analyzed separately.

raw coal cleaned and it will be noted that the skewness on the high-ash side is of a low order. The probable error decreases as size decreases. In this connection it may be noted that the average ash of the raw coal increases as size decreases, while the ash of the raw coal floating at the gravity of separation decreases as size decreases.

PLANT *C*—PITTSBURGH SEAM

This plant is handling the output of five mines in the Pittsburgh seam. Part of the coal is hand-loaded and part is machine-loaded. The Rheolaveur cleaning plant is treating 4-in. to 0-in. coal and consists of a combination of units shown in Figs. 9 and 4.

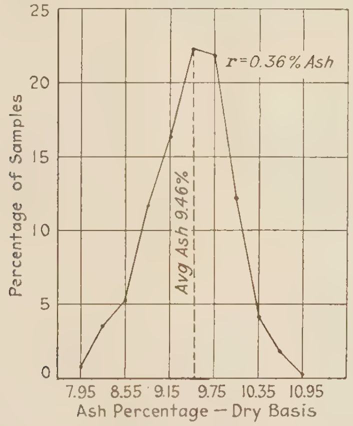

FIG. 10 DISTRIBUTION FROM AVERAGE OF WASHED $1^1/_4$-IN. TO 0-IN. SCREENINGS FROM AN ILLINOIS STRIP MINE, PLANT *B*
(Values plotted were obtained from analyses of 489 samples of cleaned screenings.)

Data are not available on all sizes of raw coal corresponding to the cleaned-coal sizes. However, such data for $1^1/_8$-in. to 0-in. and 4-in. to 0-in. raw coal are available and these are given in Table 4 along with data for cleaned coal. The cleaned coal shows a probable error of one third to one half the probable error of the raw coal, and the smallest size has the lowest such value.

PLANT *D*—PITTSBURGH SEAM

This plant is treating the output of mines working the Pittsburgh seam. The run-of-mine coal is part hand-loaded and part machine-loaded. The raw coal from 4 in. to 0 in. size,

TABLE 5 VARIABILITY OF ASH CONTENT OF 4-IN. TO 0-IN. RAW AND CLEANED COAL FROM THE PITTSBURGH SEAM, PLANT *D*

	Raw: Before bins	Raw: After bins	Cleaned
Number of cars	....	....	2410
Number of increments per car	....	....	10
Weight of each increment, lb	60	60	20
Number of increments per sample	1	1	.....
Number of samples	120	102	.....
Ash, dry basis, per cent:			
Average	10.40	10.20	6.00
Maximum	24.30	14.00	8.00
Minimum	6.70	7.80	5.00
Probable error	1.53	0.86	0.14

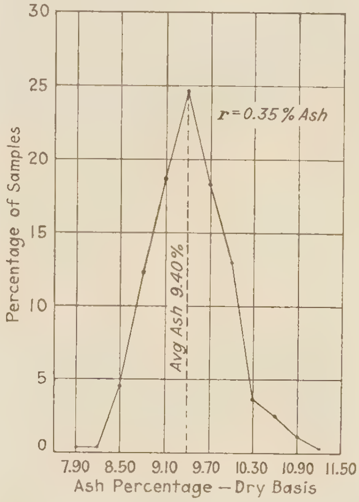

FIG. 11 DISTRIBUTION FROM AVERAGE OF WASHED $^3/_4$-IN. TO 0-IN. SCREENINGS FROM AN ILLINOIS STRIP MINE, PLANT *B*
(Values plotted were obtained from analyses of 277 samples of cleaned screenings.)

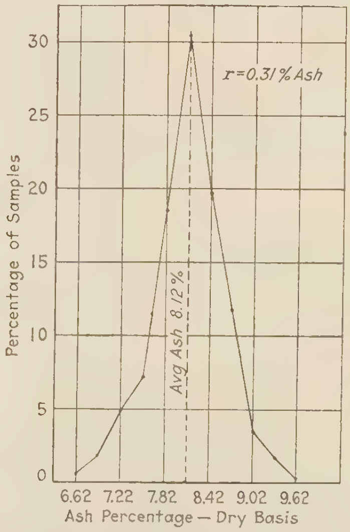

FIG. 12 DISTRIBUTION FROM AVERAGE OF WASHED $^5/_{16}$-IN. TO 0-IN. SCREENINGS FROM AN ILLINOIS STRIP MINE, PLANT *B*
(Values plotted were obtained from analyses of 367 samples of cleaned screenings.)

is cleaned to a metallurgical quality in a combination of coarse-coal and fine-coal units similar to those shown in Figs. 2 and 4, respectively. These units, however, are somewhat more elaborate and the primary refuse of the coarse-coal unit is progressively crushed during its progress through the plant.

The data on raw and cleaned coal are given in Table 5. It will

be noted that the raw-coal bins materially reduce the variability of the raw coal so that when fed to the cleaning equipment it is more uniform than all the other raw coals mentioned in this paper. Nevertheless, the cleaned coal has an unusually low variability and the probable error of the cleaned coal is less than one sixth that of the raw coal.

Conclusions

In drawing conclusions, valid comparisons cannot be made between the performance as to variability of different plants treating different coals unless the true variability of the coal itself, as well as that introduced by the methods of sampling and analysis, can be evaluated in each case. No attempt has been made by the author to determine these variability factors, and it is doubtful that they could be determined without considerable additional data. The author believes that the data submitted in this paper warrant conclusions regarding changes in variability of cleaned coal in relation to that of the raw coal which is generally shown by these data.

The cleaned-coal produced by Rheolaveur plants show the following uniformity or variability characteristics in comparison with the raw coal: (1) The uniformity is materially increased, i.e., the variability is decreased to a value one half to less than one sixth of that of the raw coal; (2) the tendency of the raw coal to skewness on the high-ash side is largely eliminated in the cleaned coal; and (3) the fine or small cleaned coal generally shows the greatest uniformity and the tendency to skewness on the high-ash side is extremely small.

A Theory of Paper Drying

By E. COWAN[1] and B. COWAN,[2] DOLBEAU, P. Q., CANADA

The authors develop theoretical formulas for predicting the performance of paper-drying machines and compare the results obtained by using these formulas with representative machines now in use. The paper includes a discussion of (1) the problems met with in the paper-making industry and how these problems can be dealt with theoretically in an effort to predict the performance of drying machines and (2) fundamental equations which can be used in designing and perfecting new equipment.

Nomenclature

The following nomenclature is used in the paper:

A_t = total surface area of a strip of drier shell 1 ft wide, sq ft
A_T = total area of cylindrical surface of drier drum, sq ft
A_u = uncovered surface area of a strip of drier shell 1 ft wide, sq ft
A_U = total uncovered surface area of a single drier drum, including the ends, sq ft
C = contact length on a drier shell, ft
c_1 = conductivity coefficient of the shell material, Btu per sq ft per deg F per in. per hr
c_2 = conductivity coefficient of the paper, Btu per sq ft per deg F per in. per hr
D = CK_p/S_pPW_p
e = weight of water evaporated from 1 sq ft of sheet for any given drier drum, lb
H = thickness of drier shell, ft
k, k_o, k_o' = surface coefficients of heat transmission, Btu per sq ft per deg F per hr
K' = coefficient of heat transmission between steam and surface of drier drum, Btu per sq ft per deg F per hr
K_p = coefficient of heat transmission through paper, Btu per sq ft per deg F per hr
l_o = latent heat of vaporization at t_o, Btu per lb
M = a constant = $(t_a - t_o)/l_o$
n and n_e = number of effective driers in a drier bank
n_p = number of preheating driers in a drier bank
P = paper thickness, ft
Q = heat input to any drum, Btu
Q_e = heat added to 1 sq ft of sheet on any given drier drum, Btu
Q_E = total heat input to n effective driers, Btu per hr
Q_p = total heat input to n_p preheating driers, Btu per hr
R = a constant = $[(t_a - t_o)/l_o]W_fS_f$
S_f = specific heat of dry fiber, Btu per lb per deg F
S_s = specific heat of shell material, Btu per lb per deg F
S_p = specific heat of paper web, Btu per lb per deg F
t_a = surface temperature of a drier shell = maximum paper temperature attained on the drier, F
t_{a1}, t_{a2}, t_{an} = surface temperature of the preheating driers, F
t_o = evaporation temperature, F
T_c = surface temperature of the drier shell at point of contact of the shell and paper, F
T_d = mean temperature difference between the steam and shell surface
T_s = surface temperature of the drier shell at point of separation of the shell and paper, F
T_1 = temperature of sheet leaving the last press, F
t_1 = steam temperature in the effective driers, F
t_1' = steam temperature in the preheating driers, F
t_p' = paper temperature on the preheating driers, F
V = paper speed, ft per hr
V' = paper speed, fpm
W = width of sheet dried, ft
W_d = total water evaporated from n driers, lb per hr
W_f = weight of bone-dry fiber per sq ft of sheet, lb
W_p = specific weight of paper web, lb per cu ft
W_s = specific weight of shell material, lb per cu ft
x_1 = thickness of drier, in.
x_2 = thickness of paper, in.
x_3 = thickness of felt shell, in.
X = maximum permissible rate of evaporation, lb of water per sq ft of sheet per hr
Y = water content of paper, lb per sq ft of sheet
Y_a = water content of sheet on the drum on which the average surface temperature is attained, lb per sq ft of sheet
Y_i and Y_f = initial and final values of Y
Z_o = free interroll space, ft
ω = angular velocity of drier drum, radians per hr
ϕ_2 and ϕ_1 = portions of drier shell in contact with paper and uncovered, respectively, radians
η = efficiency of whole drier bank

1—THEORY AND DERIVATION OF EQUATIONS

Introduction

Because of the flexibility of the usual paper-drying systems, a strict theoretical analysis of the thermodynamics involved has never been found absolutely necessary for design. It may be said that the paper machine drier has merely evolved to the extent that existing machines have always furnished the necessary criteria upon which new designs were based, and successful operation has always been possible because the different variables such as speed, steam temperature, and volume of air used could always be utilized for adjustment.

The paper machine drier, illustrated in Fig. 1 being ill adapted to experimentation, it becomes necessary to develop a rational theoretical basis for the drying process in order to achieve optimum operating conditions, and in addition, the accurate

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Contributed by the Process Industries Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers to be held at New York, N. Y., November 30 to December 4, 1936.

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prediction of the performance of new machines is highly desirable. It is for this reason that the following analysis has been made.

It should be pointed out beforehand that very few data of an experimental nature were available; the most common fault being the vague and inconclusive manner in which most investigators have described the conditions of their tests.

In making numerical calculations it has been necessary to adopt coefficients which are used under circumstances most closely related to this problem.

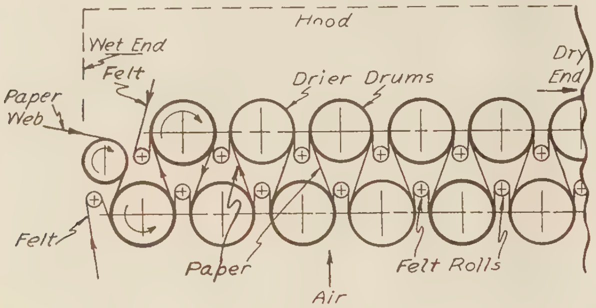

Fig. 1 Arrangement of a Drier Bank

A simple way to visualize the drying process is to consider the moist paper web as a cooling coil in contact with the hot drier. Practically all the heat of vaporization is liberated after the sheet leaves the drier; hence, the actual escape of the moisture occurs in a series of explosions. The moist sheet containing from 70 to 75 per cent by weight of moisture enters the drier nest at a temperature of from 60 to 100 F. The first two or three driers heat the moist sheet to the evaporation temperature. The sheet is held in close contact with the shell by means of an endless felt. Heat added to the sheet on the remaining driers, referred to as effective driers in subsequent calculations, is used to evaporate the moisture. The derivation of the formulas in this study is based on the following hypothesis describing the drying cycle on a given drier.

The thin moist paper web enters the drier at the evaporation temperature t_o. It is assumed that very little heat, and in turn moisture, escapes during the contact part of the cycle. Before leaving the drier roll the moist sheet has acquired a temperature approximating that of the surface of the drier shell. On breaking contact, the sheet begins to cool and evaporation of the entrained moisture takes place. The rate of evaporation is proportional to the difference in vapor pressure between the vapor in the sheet and the vapor in the air in the immediate vicinity of the sheet. It is further assumed that sufficient air at a constant temperature and relative humidity is being supplied to carry away the moisture. The longer the time interval between driers, the greater will be the cooling of the sheet and, hence, the lower the average evaporation temperature. Given the evaporation temperature at a certain speed, the corresponding evaporation temperature t_o may be calculated for other speeds. As soon as the wet paper leaves the shell, the shell tends to become hotter, thus acting as a reservoir for heat that is released when contact is made again. While this difference is only a fraction of a degree, the heat thus stored is of considerable magnitude due to the number of cycles per unit of time. The rate at which water is evaporated must be such that the steam in the sheet escapes without injuring the formation. The maximum permissible rate is governed by the strength or cohesion of the fiber mass necessary to prevent distortion and by the hygroscopic characteristics of the sheet, governed in turn by the freeness, the percentage of sulphite pulp, and the formation of the sheet. This is largely a paper-making, not an engineering, problem and, as the factors are not quantitatively known, the authors have endeavored to choose conditions within the range of known practice. Ignorance of these factors does not affect the validity of the equations given in the paper.

The total pressure of the air-vapor mixture in the drier hood is composed of dry-air pressure plus vapor pressure, each of these being a partial pressure obeying Dalton's law. Doubtless there is a considerable range in evaporation temperatures for any given condition. We are concerned only with an average evaporation temperature. The equations show the relation between all of the important variables.

Scavenging air receives its additional heat from (1) heat recovery through a heat interchanger using the exhaust air-vapor mixture; (2) steam coils when low atmospheric temperatures prevail; (3) the vapor from the sheet which in cooling surrenders some of its sensible heat and causes the dry-bulb temperature of the air to increase; and (4) radiation and convection from the driers.

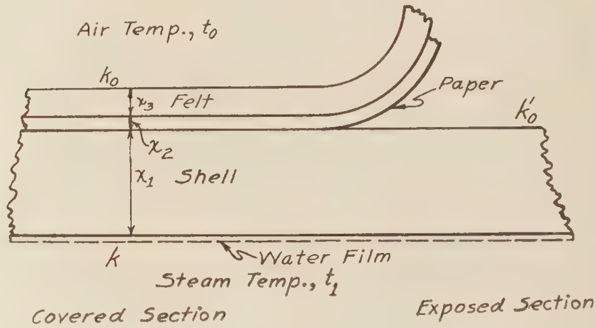

Fig. 2

Air as it leaves the driers will be at about 140 F and have a relative humidity of from 40 to 50 per cent.

Radiation from the hood itself may cool this air to around 120 F and increase its relative humidity to 70 or 85 per cent. For the maximum heat recovery, it is obvious that the temperature of the exhaust vapor must be high. It is probable that relative humidities greater than 60 per cent at 140 F retard the rate of evaporation. For a specific example, there are limits to the temperature of the air supply below which the drying capacity is reduced and above which there is no appreciable increase in the drying rate.

Calculation of Drier Surface Temperature

Consider first the properties of a single drier drum. In all of the following calculations it will be assumed that all surfaces are flat planes, and the effect of the curvature of the drier shell on the heat transmission can be neglected.

Referring to Fig. 2, let x_1, x_2, and x_3 = thicknesses of shell, paper, and felt, respectively, in.; c_1, c_2, and c_3 = corresponding conductivity coefficients, Btu per sq ft per deg F per hr per in.; and k, k_o, k_o' = surface coefficients, Btu per sq ft per deg F per hr.

Consider here that the air temperature t_o in the immediate vicinity of the drier shell is the temperature at which evaporation from the sheet takes place.

While the average air temperature in the drier hood is much lower than the actual evaporation temperature t_o, it is a fact confirmed by personal observation that comparatively high air temperatures are obtained close to the sheet, while the air at a distance is much cooler. It may also be shown that most of the heat added to the air-vapor mixture in the drier bank is due to the heat of evaporation of the water. It appears certain, therefore, that in the immediate vicinity of the sheet, the air temperature must be approximately the same as the evaporation

temperature. For the sake of simplicity this temperature has also been assumed to exist in the layer close to the uncovered portion of the drier shell. Whether or not this latter assumption is strictly true is a matter of little importance, since the heat quantities involved are relatively small.

The coefficient of heat transmission between the steam and the surface of the shell is

$$K' = \frac{1}{(1/k) + (x_1/c_1)}$$

where K' is the coefficient of heat transmission, Btu per sq ft per deg F per hr.

Consider a portion of the shell during a single revolution. Let the surface temperature of the shell be T_c at the point of contact of the shell and paper. As the paper becomes heated, its temperature rises from t_o to t_a, and the temperature of the shell surface falls from T_c to T_e. During rotation across the uncovered portion, the surface temperature rises again from T_e to T_c.

Consider the uncovered portion of the shell, and an element of surface area dA, moving through an angle $d\theta$ in time ds as shown in Fig. 3. The surface temperature of the shell at θ is T, and at $\theta + d\theta$ is $T + dT$. Then the heat transmitted through the shell in the time ds is equal to the heat radiated to the air

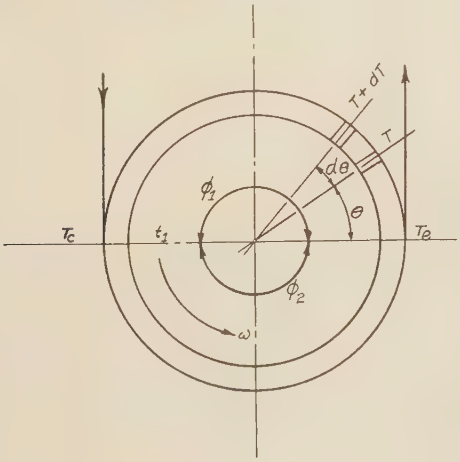

FIG. 3

plus the heat absorbed in raising the shell temperature by an amount dT. The heat transmitted through the shell is

$$K'(t_1 - T)dAds$$

and the heat lost to the air is

$$k_o'(T - t_o)dAds$$

If H = thickness of shell, ft; W_s = specific weight of shell material, lb per cu ft; and S_s = specific heat of shell material, Btu per lb per deg F; then, the heat absorbed in raising the shell temperature is

$$HW_sS_sdAdT$$

and

$$K'(t_1 - T)dAds = k_o'(T - t_o)dAds + HW_sS_sdAdT$$

or

$$ds[(K't_1 + k_o't_o) - T(K' + k_o')] = HW_sS_sdT$$

If the angular velocity of the drum is ω radians per hr, so that $ds = d\theta/\omega$, then

$$\frac{d\theta}{\omega HW_sS_s} = \frac{dT}{(K't_1 + k_o't_o) - T(K' + k_o')}$$

Integrating T from T_c to T_e, and θ from 0 to ϕ_2 gives the total change in temperature, that is

$$\int_{T_c}^{T_e} \frac{dT}{(K't_1 + k_o't_o) - T(K' + k_o')} = \int_0^{\phi_2} \frac{d\theta}{\omega HW_sS_s}$$

and

$$\frac{1}{K' + k_o'} \log_e \left[\frac{(K't_1 + k_o't_o) - T_c(K' + k_o')}{(K't_1 + k_o't_o) - T_e(K' + k_o')}\right] = \frac{\phi_2}{\omega HW_sS_s} \quad \text{........[1]}$$

A solution of Equation [1] using actual values and known data shows the relation between T_c and T_e. Consider a 60-in. drier using steam at 10 lb per sq in. gage, and operating at a speed of 1200 fpm. From Fig. 6 it is observed that the evaporation temperature t_o at this speed is 180 F. Also, t_1 = 240 F, k = 2130 Btu per sq ft per deg F per hr,[3] x_1 = 1.125 in., and c_1 = 313.4 Btu per sq ft per hr per deg F per in.,[4] so that K' = 246.5 Btu per sq ft per deg F per hr. Based on a surface coefficient of 1.34 and a windage factor[5] of 4.5, k_o' = 6.1. Other values are: ω = $[(1200/5\pi) \times 2\pi \times 60]$ = 28,800 radians per hr, $H = x_1/12$ = 0.0938 ft, W_s = 480 lb per cu ft, S_s = 0.1189 Btu per lb per deg F,[6] and approximately, $\phi_2 = \pi$.

Substituting the foregoing values in Equation [1] it is found that

$$\frac{238.5 - T_c}{238.5 - T_e} = 0.995$$

For a value of T_c = 200, T_e = 199.6 F. Then T_c and T_e are very nearly equal.

Now consider the covered portion of the shell. The heat transmitted through the shell is the difference between the heat carried away by the paper and the heat rejected by the shell. The heat transmitted through the shell during this part of the cycle = $K'T_d(\phi_1/\omega)$ where T_d is the mean temperature difference between the steam and the shell surface. Since T_c and T_e are very nearly equal

$$T_d = [t_1 - (T_c + T_e)/2]$$

If Y = the water content, lb per sq ft of paper on the drier drum; W_f = weight of bone-dry fiber per sq ft of sheet; and S_f = specific heat of fiber, Btu per lb per deg F, then the heat absorbed by 1 sq ft of paper in its passage over the drum is

$$(Y + W_fS_f)\ (t_a - t_o)$$

But, since the paper and shell are held in very close contact, it may be assumed that $t_a = T_c$.

The heat rejected by the shell per square foot of surface is

$$HW_sS_s(T_c - T_e)$$

Then the heat transmitted through the shell is

$$K'(\phi_1/\omega)[t_1 - (T_c + T_e)/2] = (Y + W_fS_f)(T_c - t_o) - HW_sS_s\ (T_c -$$

[3] Kent's Mechanical Engineers' Handbook, p. 608.
[4] Ibid., p. 602.
[5] Ibid., p. 634.
[6] Ibid., p. 592.

or

$$T_c[(Y + W_fS_f) - HW_sS_s + K'(\phi_1/2\omega)] + T_e[K'\phi_1/2\omega + HW_sS_s] = K'(\phi_1/\omega)t_1 + (Y + W_fS_f)t_o \qquad [2]$$

By solving Equations [1] and [2] for the two unknowns, the actual values of T_c and T_e can be determined. The differential, however, is so small that large errors in calculation are difficult to avoid. By considering $T_c = T_e = t_a$, a simpler solution is obtained.

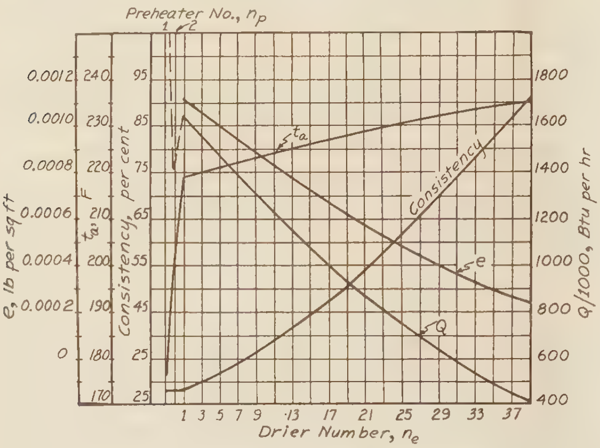

FIG. 4 THEORETICAL DRIER PERFORMANCE

(The curves are based on the following conditions: Diameter of drier drums = 60 in.; steam pressure in effective driers = 10 lb per sq in. gage; steam pressure in preheater driers = 2 lb per sq in. gage; initial consistency of paper = 28 per cent; final consistency of paper = 92 per cent; paper speed = 1200 fpm; weight of paper = 32 lb per 3000 sq ft of air-dry paper.)

Consider a strip of shell of unit width and let V = surface speed, ft per hr; A_t = total surface area of the shell, sq ft; and A_u = area of the uncovered surface, sq ft. Then, the total heat transmitted from the inside of the shell to the surface, which is the sum of the heat carried away by the paper and the heat radiated to the air from the uncovered surface, can be expressed as

$$K'A_t(t_1 - t_a) = V(Y + W_fS_f)(t_a - t_o) + k_o'A_u(t_a - t_o)$$

from which

$$t_a = \frac{K'A_tt_1 + Vt_o(Y + W_fS_f) + k_o'A_ut_o}{K'A_t + V(Y + W_fS_f) + k_o'A_u} \qquad [3]$$

For any given speed and moisture content of the paper, the surface temperature of any drier shell can be obtained with Equation [3]. Fig. 4, which was plotted from the values given in Table 1, shows the variation in t_a, for the specified conditions over the whole of a given drier bank. The variations in temperature, over the effective driers, follow almost a straight line, and the arithmetic mean of the initial and final temperature is close to the actual mean value. While it is possible to determine the characteristics of a drier bank through a process of arithmetic integration, such as has been performed in Table 1, this method is cumbersome. It has been found sufficiently accurate to consider a representative, or mean, drier, the characteristics of which are an average for the whole drier bank. Thus, if t_{ai} = the surface temperature of the first effective drier, where $Y = Y_i$, and t_{af} = the surface temperature of the last drier, where $Y = Y_f$, the average temperature

$$t_a = \frac{t_{ai} + t_{af}}{2} \qquad [4]$$

In a given machine, operating with a fixed number of drier

TABLE 1 PERFORMANCE DATA OF A PAPER-DRYING MACHINE

Drier no.	Y, lb per sq ft	Consistency, %	t_a, F	e, lb per sq ft	Q, Btu per hr
1	0.0252	28.0	219.0	0.00112	1,645,000
2	0.0241	28.9	219.5	0.00109	1,605,000
3	0.0230	29.9	220.0	0.00106	1,561,000
4	0.0219	30.9	220.5	0.00103	1,518,000
5	0.0209	31.9	221.0	0.00100	1,477,000
6	0.0199	33.0	221.5	0.000972	1,435,000
7	0.0189	34.2	222.1	0.000945	1,397,000
8	0.0180	35.3	222.6	0.000916	1,355,000
9	0.0171	36.5	223.1	0.000887	1,316,000
10	0.0162	37.7	223.5	0.000856	1,271,000
11	0.0153	39.1	224.0	0.000827	1,231,000
12	0.0145	40.3	224.5	0.000800	1,193,000
13	0.0137	41.7	225.0	0.000773	1,154,000
14	0.0129	43.2	225.5	0.000746	1,115,000
15	0.0122	44.5	226.0	0.000720	1,082,000
16	0.0115	46.1	226.5	0.000694	1,044,000
17	0.0108	47.8	227.0	0.000668	1,007,000
18	0.0101	49.2	227.5	0.000643	972,000
19	0.00944	50.9	228.0	0.000618	938,000
20	0.00883	52.6	228.5	0.000595	905,000
21	0.00823	54.4	229.0	0.000572	873,000
22	0.00766	56.1	229.5	0.000549	841,000
23	0.00711	57.9	230.0	0.000526	809,000
24	0.00658	59.9	230.0	0.000500	772,000
25	0.00608	61.7	231.0	0.000493	750,000
26	0.00559	63.7	231.5	0.000463	721,000
27	0.00513	65.6	231.5	0.000439	687,000
28	0.00469	67.7	232.0	0.000420	661,000
29	0.00427	69.6	232.0	0.000399	629,000
30	0.00387	71.7	233.0	0.000385	611,000
31	0.00348	73.9	233.0	0.000364	580,000
32	0.00312	75.8	233.0	0.000345	553,000
33	0.00278	78.0	233.5	0.000330	532,000
34	0.00245	80.0	233.8	0.000314	509,000
35	0.00213	82.1	234.0	0.000297	487,000
36	0.00184	84.1	234.3	0.000283	466,000
37	0.00155	86.4	234.5	0.000269	446,000
38	0.00128	88.4	234.8	0.000255	427,000
39	0.00103	90.5	234.9	0.000242	408,000
		92.5			

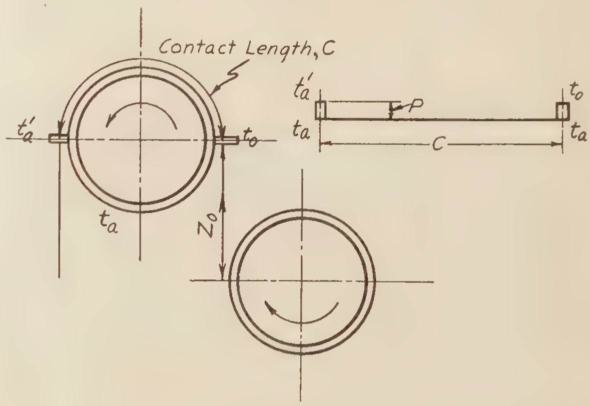

FIG. 5

drums, any variation in speed must be met with changes in the temperature of the drums if the final dryness of the product is to be maintained the same. To effect this, the steam pressure in the drums is changed. Consider, as in the previous example, a 60-in. drier with a shell thickness of 1.125 in. The usual values for the initial and final consistency of the sheet are 28 and 92 per cent, respectively. Using these values, the calculation of the average surface temperature from Equations [3] and [4] shows that an empirical relationship between t_a, t, and V can be obtained which is strictly accurate between the limits set. Thus for the foregoing conditions, the average surface temperature t_a can be represented as

$$t_a = (0.9552 - 0.000144V')t_1 + 0.0314V' + 1.48$$

and for a 48-in. drier

$$t_a = (0.942 - 0.0001616V')t_1 + 0.0354V' + 2.81$$

where V' = speed, fpm. The two equations just given can be used at all speeds between 600 and 1200 fpm, and for values of t_1 usually encountered.

Determination of Paper Temperature

In its passage over the drum, the paper is held in firm contact with the surface of the shell, and the surface resistance to heat transmission is negligible. Also, since the paper is very thin (between 0.003 and 0.007 in.), the average temperature of the paper may be considered as its surface temperature.

Referring to Fig. 5, let C = contact length on the drum, ft; P = thickness of the paper, ft; t_a' = temperature at which the paper leaves the shell surface, F; K_p = coefficient of heat transmission through the paper, Btu per sq ft per deg F per hr; S_p = specific heat of the paper, Btu per lb per deg F per hr; W_p = specific weight of the paper, lb per cu ft. Consider an element of paper of surface area dA. If the felt is considered as a perfect insulator, all of the heat transmitted from the drum to the paper is used in raising the paper temperature from t_o to t_a'. The heat transmitted from the shell surface to the surface of the paper is

$$K_p \frac{t_a' - t_o}{\log_e \dfrac{t_a - t_o}{t_a - t_a'}} dA \frac{C}{V}$$

and the heat absorbed in raising the temperature of the paper is

$$(t_a' - t_o)\, S_p\, PdAW_p$$

Therefore

$$K_p \frac{t_a' - t_o}{\log_e \dfrac{t_a - t_o}{t_a - t_a'}} dA \frac{C}{V} = (t_a' - t_o) S_p PdAW_p$$

and

$$\log_e \frac{t_a - t_o}{t_a - t_a'} = \frac{CK_p}{S_p P W_p V} \qquad [5]$$

Consider a speed of 1200 fpm. From Table 1 it is seen that on the twenty-fourth drier the paper has a composition of approximately 60 per cent dry fiber and 40 per cent water, and that t_a = 230 F. The evaporation temperature t_o is 180 F. For a 60-in. drier, C = 7.85 ft, approximately. The paper thickness = (0.007/12) ft, the specific weight W_p = 28 lb per cu ft, the thermal conductivity of water[7] = 4, and the thermal conductivity of oak, which may be considered a reasonable value for dry fiber,[8] = 2.5. Using these values, the composite thermal conductivity of the paper $c_2 = (4 \times 0.4) + (2.5 \times 0.6) = 3.1$, and the coefficient of heat transmission through the paper $K_p = c_2/x_p$ = 3.1/0.007 = 442.8 Btu per sq ft per deg F per hr.

Similarly, the specific heat of dry fiber S_f = 0.34 Btu per lb per deg F, and the composite specific heat = $(0.4 \times 1) + (0.6 \times 0.34)$ = 0.604 Btu per lb per deg F.

Substituting these values in Equation [5]

$$\log_e \frac{230 - 180}{230 - t_a'} = 4.89$$

or t_a' = 229.6 F, that is, with the driers in actual use, there is always sufficient time for the paper to attain very closely the temperature of the shell surface. In the calculations to follow, it will now be considered that the final (maximum) temperature of the paper is equal to the temperature of the drum surface.

Drying Performance of a Complete Drier Bank

Consider any square foot of paper passing over a drier drum. The heat absorbed by the paper in its passage over the drum is the sum of the heat utilized in evaporating moisture from the paper and the heat radiated to the air in the interroll space. The heat radiated to the air in the interroll space is small enough to be neglected. Then, if e = weight of water evaporated from 1 sq ft of sheet for any given drum, lb; and l_o = latent heat of vaporization at the evaporation temperature t_o, Btu per lb; then, the heat absorbed by the paper in its passage over the drum is

$$Q_e = (t_a - t_o)\,(Y + W_f S_f)$$

and

$$e = \frac{(t_a - t_o)\,(Y + W_f S_f)}{l_o} \qquad [6]$$

If Y_i = the initial water content of the paper entering the drier bank, lb per sq ft; Y_f = final water content of the paper leaving the drier bank, lb per sq ft; Y_2 = water content of the paper on No. 2 drier, lb per sq ft; and Y_n = water content of the paper on the nth drier, lb per sq ft; then, the water evaporated from the paper on the first effective drier is

$$e_1 = \frac{(t_a - t_o)\,(Y_i + W_f S_f)}{l_o}$$

and

$$Y_2 = Y_i - e_1 = Y_i - [(t_a - t_o)/l_o]Y_i - [(t_a - t_o)/l_o]W_f S_f$$

Since we have assumed an average value for t_a, it can be said that

$$(t_a - t_o)/l_o = \text{a constant} = M$$

and

$$MW_f S_f = \text{a constant} = R$$

Then

$$Y_2 = Y_i(1 - M) - R$$

On drier No. 2, the water evaporated is

$$e_2 = \frac{(t_a - t_o)}{l_o}(Y_2 + W_f S_f) = MY_2 + R$$

The weight of water in the sheet leaving the second drier, after complete evaporation has taken place, is

$$Y_3 = Y_2 - e_2 = Y_2(1 - M) - R = Y_i(1 - M)^2 - R(1 - M) - R$$

Similarly, the water in the sheet leaving the nth drier is

$$Y_{n+1} = Y_i(1 - M)^n - R(1 - M)^{n-1} - R(1 - M)^{n-2} + \ldots\ldots - R(1 - M) - R \qquad [7]$$

But, if n = number of effective driers in the drier bank, then $Y_{n+1} = Y_f$, so that

$$Y_f = Y_i(1 - M)^n - R[(1 - M)^{n-1} + (1 - M)^{n-2} + \ldots\ldots + (1 - M) + 1] \qquad [8]$$

If Equation [8] is multiplied by $(1 - M)$ and subtracted from Equation [7], then

$$Y_f M = -Y_i(1 - M)^{n+1} + Y_i(1 - M)^n + R(1 - M)^n - R = (1 - M)^n (Y_i M + R) - R$$

from which

$$(1 - M)^n = \frac{Y_f M + R}{Y_i M + R}$$

and

$$n = \frac{\log(Y_f M + R) - \log(Y_i M + R)}{\log(1 - M)}$$

[7] Kent's Mechanical Engineers' Handbook p. 602.

[8] Ibid., p. 630.

Substituting for M and R and simplifying

$$n = \frac{\log(Y_f + W_f S_f) - \log(Y_i + W_f S_f)}{\log(l_o + t_o - t_a) - \log l_o} \quad [9]$$

Thus, the number of effective driers required can be calculated from Equation [9] if the initial and final moisture content of the sheet, the steam temperature in the shell, and evaporation temperature t_o are given.

Evaporation Temperatures

The evaporation temperature is made a function of the machine speed only, and its values may be determined as follows:

At a speed of 1200 fpm the average evaporation temperature is 180 F. This value has been determined from the performance data of actual machines, and is assumed as a datum from which

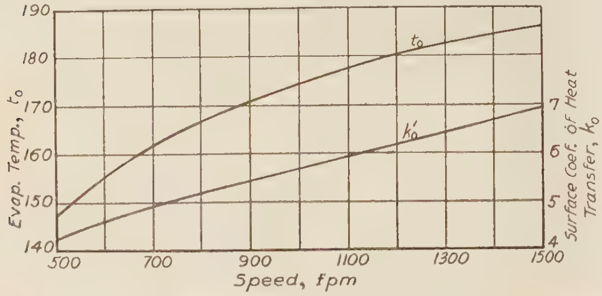

Fig. 6 Surface Coefficients and Evaporation Temperatures at Various Paper Speeds

to calculate evaporation temperatures at other speeds. It has also been demonstrated that the rate of evaporation increases directly as the air velocity. Carrier[9] gives the factor $1 + (V'/230)$. The greater the time interval between driers, the greater will be the drop in temperature of the sheet, and the lower will be the evaporation temperature. Then

$$\frac{t_o \text{ at } 1200 \text{ fpm}}{t_o \text{ at } V' \text{ fpm}} = \frac{1 + (V'/230)}{1 + (1200/230)} \, \frac{1200}{V'}$$

and the required evaporation temperature is

$$t_o = \frac{180V'}{193 + 0.84V'} \quad [10]$$

These values are plotted in Fig. 6. Therefore, values of the evaporation temperature within the usual range can be obtained directly from Fig. 6. It is to be noted that the calculated evaporation temperature is an average temperature. There will be an actual gradient, ranging from a high value of the evaporation temperature at the instant the sheet leaves the drum, to successively lower values as the sheet cools.

The evaporation temperature is considerably higher than the temperature of the air leaving the hood. The water vapor will therefore tend to cool, losing some of its sensible heat to the air, and the actual heat content of the vapor leaving the drier nest will be the sum of the total heat corresponding to the partial pressure of the vapor and the heat in the dry air. The possibility that this vapor contains some superheat may be neglected.

It should be added that the calculated values of the evaporation temperatures are based on an incoming air temperature of 110 F, which is the temperature usually found in practice.

Preheating Driers

As stated previously, the paper leaves the last press at a temperature between 60 and 100 F, and is heated to the evaporation temperature t_o before evaporation of the moisture begins. The driers required for this purpose are called the preheating driers.

Let T_1 = temperature of the paper leaving the last press, F; t_{p1} = final temperature of the paper leaving preheater drier No. 1; and t_1' = steam temperature in the preheating driers. The steam temperature t_1' is usually lower than t_1 because of the fact that rapid heating of the wet sheet causes the fibers to stick to the drum, and also produces a wrinkling of the paper; for this reason, flash steam from the condensate removed from the effective driers is used in the preheating driers. Then, from Equation [3]

$$t_{a1} = \frac{K'A_t t_1' - V(Y_i + W_f S_f)(t_{p1} - T_1) + k_o' A_u t_o}{K'A_t + k_o' A_u} \quad [11]$$

From Equation [5], the actual paper temperature leaving drier No. 1 may be determined as follows:

$$\log_e \frac{t_{a1} - T_1}{t_{a1} - t_{p1}} = \frac{CK_p}{S_p P W_p V}$$

For a given set of conditions

$$\frac{CK_p}{S_p P W_p} = \text{a constant} = D$$

Then

$$\frac{t_{a1} - T_1}{t_{a1} - t_{p1}} = e^{D/V}$$

and

$$t_{p1} = \frac{t_{a1}(e^{D/V} - 1) + T_1}{e^{D/V}} \quad [12]$$

Equations [11] and [12] can be solved for t_p.

The number of preheating driers necessary can now be calculated by determining the drier on which t_{pn} is equal or greater than t_o. From this point on, evaporation takes place. For example, take t_1' = 220 F (2 lb per sq in. gage), T_1 = 80 F, speed = 1200 fpm, and 60-in. driers. From Equations [11] and [12] it is found that t_{a1} = 176.5 F and t_{p1} = 162.5 F. Similarly, t_{a2} = 201.5 F and t_{p2} = 197 F. In this case, t_o = 180 F, and therefore only two preheating driers are necessary.

Drying Efficiency

The total heat input to a drier drum is the sum of the heat carried away by the moist paper and the heat added to the air by radiation and convection from the uncovered surface. If A_U = total uncovered surface of a single drum, including the ends, sq ft; W = width of dried sheet, ft; and Y_a = water content of the sheet corresponding to the average temperature t_a, lb per sq ft; then, the heat input to any drum is

$$Q = k_o'(t_a - t_o)A_U + (t_a - t_o)(Y + W_f S_f)VW \quad [13]$$

A curve giving the heat input Q to any drum in a complete drier bank is shown in Fig. 4, from which it may be seen that the drum on which the average surface temperature is obtained will also give approximately the average heat consumption. In this case, the total heat consumption for n_e effective driers is approximately

$$Q_E = Qn_e = n_e(t_a - t_o)[k_o' A_U + VW(Y_a + W_f S_f)] \quad [14]$$

The heat input to the preheating driers may be approximated by considering only the heat used in raising the paper temperature from T_1 to t_o, and neglecting radiation to the air. In this case the total heat input to the preheater drums is

$$Q_p = (t_o - T_1)(Y_i + W_f S_f)VW \quad [15]$$

[9] Kent's Mechanical Engineers' Handbook p. 1448.

The efficiency of the complete drier bank is

$$\eta = \frac{\text{heat used in evaporating water}}{\text{total heat input}} = \frac{W_d\, l_o}{Q_E + Q_p} \ldots [16]$$

Rate of Drying

It was stated previously that there is a maximum rate at which the moisture in the sheet can be evaporated without injury to the sheet due to steam bubbles. If X = maximum permissible rate of evaporation, lb of water per sq ft of sheet per hr, the free interroll space Z_o, as shown in Fig. 5, must be such as to allow the water actually evaporated to leave at a rate equal to, or less than, X. But

$$\frac{Z_o}{V} X = \frac{(t_a - t_o)(Y + W_f S_f)}{l_o}$$

so that

$$Z_o = \frac{V(t_a - t_o)(Y + W_f S_f)}{X\, l_o} \ldots\ldots\ldots\ldots [17]$$

By substituting for known conditions in Equation [17], the minimum drier spacing which allows sufficient time for the escape of the steam formed, can be determined.

2—PRACTICAL APPLICATION OF THE DRYING FORMULAS

In the manufacture of newsprint, the fixed charge is today the largest single cost. Modern newsprint machines operate at speeds ranging from 900 to 1400 fpm but greater speeds will doubtless be realized in the future. By reason of the uniformity of the product, the conditions of operation can be fixed, thus permitting the machine designer to set definite limits to the drying problem. The paper machine itself is the costliest piece

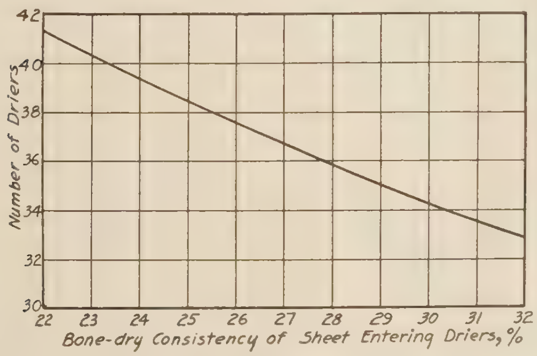

Fig. 7 Influence of Paper Wetness on the Theoretical Number of Driers

(Speed = 1200 fpm; drum diameter = 60 in.; final consistency = 92 per cent; and weight of paper = 32 lb per 3000 sq ft of air-dry paper.)

of equipment in the mill. The authors' experience has been largely with newsprint; hence, the following calculations and data are solely for a sheet consisting of about 10 to 20 per cent chemical pulp and remainder ground wood pulp. The weight of the sheet (32 lb) is expressed as the weight of 3000 sq ft of air-dry paper. The finished sheet is about 0.003 in. thick. In order to dry out wet streaks, the tendency is to overdry the paper and then permit the dried sheet to regain moisture by passing over a shell known as the "sweat drier" through which cold water is circulated. This fact is ignored in the following charts.

Fig. 4 illustrates theoretical drier performance. Loose felts, poor drainage of condensate can, and do, alter the actual performance of each drier. From Equation [9], the number of driers was first calculated. Referring to Fig. 4, ordinates from the number of the drier to each curve represent the average conditions on that drier. The surface temperature curve t_a is calculated from Equations [3] and [4]. The heat-unit curve Q shows the heat transmitted by each drum, calculated from Equation [13].

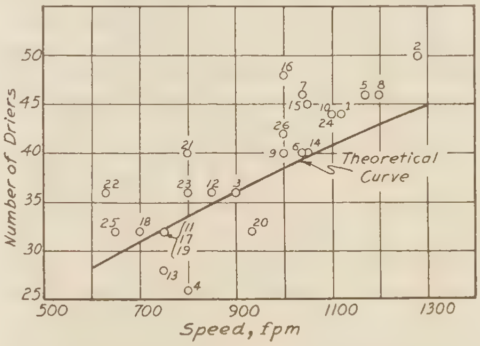

Fig. 8 Actual Number of Driers on Representative Machines Compared With Theoretical Number of Driers Computed by Authors' Equations

(Theoretical curve based on the following conditions: Diameter of driers = 60 in.; steam pressure in drums = 10 lb per sq in. gage; initial consistency of paper = 28 per cent; final consistency of paper = 92 per cent; number of preheating driers = 2; number of effective driers = n_e; total number of driers = $n_p + n_e$ + 1 sweat drier; weight of paper = 32 lb per 3000 sq ft of air-dry paper.)

The water evaporated on each drum is given by curve e, calculated from Equation [6]. The consistency curve C indicates the mean percentage by weight of bone-dry fiber in the sheet on each drier. For specific design, this curve can be assumed, and the steam supply and drainage system designed to give the required shell surface temperatures which will conform with the assumed curve. Fig. 7 shows a curve the abscissas of which represent

TABLE 2 ACTUAL NUMBER OF DRIERS IN VARIOUS MACHINES COMPARED WITH THE NUMBER OF DRIERS CALCULATED BY AUTHORS' FORMULAS

Mill no.	Drier diam., in.	Speed, fpm	Actual no. of driers	Theoretical no. of driers[a] n_e	n_p	n
1	60	1120	44	37.8	2	41
2	60	1280	50	41.7	2	45
3	60	900	36	32.7	2	36
4	60	800	26	30.6	2	34
5	60	1170	46	39.2	2	43
6	60	1040	40	36.1	2	40
7	60	1040	46	36.1	2	40
8	60	1200	46	39.5	2	43
9	60	1000	40	35.1	2	39
10	60	1100	44	37.4	2	41
11	60	750	32	29.7	2	33
12	60	850	36	31.0	2	34
13	60	750	28	29.6	2	33
14	60	1050	40	35.9	2	39
15	60	1050	45	35.9	2	39
16	60	1000	48	35.1	2	39
17	60	750	32	31.9	2	35
18	60	700	32	27.7	2	31
19	60	750	32	31.9	2	35
20	60	935	32	33.25	2	37
21	60	800	40	31.0	2	34
22	60	630	36	25.9	2	29
23	60	800	36	31.0	2	34
24	60	1100	44	37.4	2	41
25	60	650	32	26.6	2	30
26	60	1000	42	35.1	2	39
27	48	675	32	28.0	2	31
28	48	600	30	25.9	2	29
29	48	650	32	27.3	2	31
30	48	775	32	31.0	2	34
31	48	600	32	25.9	2	29
32	48	600	28	25.9	2	29
33	48	630	28	26.7	2	30
34	48	725	40	29.7	2	33
35	48	640	30	27.25	2	31
36	48	745	40	30.5	2	34
37	48	580	28	25.3	2	29
38	48	825	32	32.1	2	36
39	48	725	32	29.3	2	33

[a] $n = n_e + n_p$ + 1 sweat drier.

Note: Operating conditions of all machines: Steam temperature in effective driers, t_1 = 240 F; initial consistency of paper = 28 per cent, final consistency of paper = 92 per cent; weight of paper = 32 lb per 3000 sq ft of air-dry paper (newsprint).

increasing increments of wetness of sheet entering the driers; the ordinates are the number of drier cylinders required.

Table 2 gives the drier diameter, operating speed, theoretical and actual number of driers, for a number of representative newsprint machines operating under the following conditions: An average steam temperature of 240 F, a newsprint of 32 lb basis weight, initial bone-dry consistency of 28 per cent, and final consistency of 92 per cent. These data are used in calculating the values given in the column for n_e. The actual and theoretical number of driers have been plotted in Fig. 8 with the known operating speeds as a base. It will be noted that above 1000 fpm the actual number of driers are all above the line, indicating that the drying capacity is excessive or that the speeds for which the machines were designed have not been attained. The slow-speed machines on the other hand appear to have a speed greater than the original design speed. However, in the case of machines using exhaust steam for drying, a low back pressure may account for the larger number of driers. Fig. 8 would appear to amply demonstrate the validity of the equations presented.

These equations may also be used for vacuum drying. For example, given a speed of 1000 fpm, a drier diameter of 60 in., a 27 in. vacuum, initial and final consistencies of 28 per cent and 92 per cent, respectively, and a newsprint sheet. Vapor will leave the sheet at a temperature of 114 F. Assume a steam temperature of 240 F. By calculation, t_a = 220.6 F. Substituting for known values in Equation [9], the number of driers is equal to 18 plus 1 preheating drier. An actual installation has twenty-two 60-in. driers.

It is probable that there is some relation between the drum diameter and the speed, other things being equal. Equation [5] can be used to plot the temperature gradient of the wet sheet on a given drier. It is conceivable that with a large drier at slow speed the incremental gain of heat by the wet sheet toward the end of the contact cycle may fall so rapidly that the last 2 or 3 ft of contact length are practically sterile. Perhaps a greater drying capacity per unit area may be realized by combining a number of large and small driers. In actual practice, however, the design of the drying system should be correlated with the heat recovery from the air used to carry away the moisture as well as the design of the hood over the drier section.

A similar analysis has been made for pulp-drying machines and the calculated data obtained agree in a gratifying manner with known operating conditions. It is hoped that this theoretical outline may serve as a stimulus and perhaps as a guide to those who have at their disposal adequate facilities for experimental work.

The Thermal Conductivity of Liquids

By J. F. DOWNIE SMITH,[1] CAMBRIDGE, MASS.

New data on the thermal conductivities of several liquids are presented in this paper. A compilation of these values together with those of previous experimenters yields a large list of liquids the thermal conductivities of which have been determined. This list should be of interest to many engineers and physicists. An analysis of theoretical and empirical equations for predicting thermal conductivities of liquids is included in the paper.

It HAS been evident for some time that a satisfactory check of any equations developed for predicting the thermal conductivity of a liquid was not possible, mainly because of the lack of a sufficient number of reliable experimental data. For this reason, a large number of liquids have been tested experimentally in apparatus described previously (1, 2).[2] This consisted of two concentric copper cylinders, separated at the ends by thin German silver, with an annular space $^1/_{64}$ in. thick and $1^1/_4$ in. long, between them. The liquid to be tested is passed into this space. A heating wire through the center of the inner copper cylinder supplies the required heat, and the temperature difference of the two copper cylinders is obtained by means of thermocouples. The temperature difference is ordinarily about 0.6 C, so that convection is negligible. The author made a few changes in the apparatus to permit greater accuracy. These changes included the installation of a vacuum-tube relay in the thermostat circuit to minimize fouling of the mercury and permit closer temperature control; and a reduction in the number of contacts in the electrical circuits to reduce variable resistances and thermal effects. During tests the equipment operated remarkably well. For example, it was possible to check the temperature difference of the two junctions of the thermocouple to one fiftieth of 1 C for 15 min at a time, and during a run of perhaps 3 min, a variation of less than one hundredth of 1 C was common.

The liquids tested were chosen for several reasons. One reason was that few conductivities had ever been obtained for liquids for which the velocity of sound has been determined, thus making it difficult to check the formulas which involve this factor. Another reason was to complete a series already started by the author and others. A third reason was to supply data of interest to chemical and mechanical engineers, and to physicists.

Thermal Conductivity of Paraffin and Benzene Series, Other Liquids

The two main series tested were the paraffin series and a benzene series. The complete data are given in Table 1. The n-heptyl alcohol was tested in order to see that the equipment was operating properly, and the check with the values of Daniloff (3) is considered satisfactory. The footnotes in Table 1 tell where the various liquids were obtained.

TABLE 1 NEWLY DETERMINED THERMAL CONDUCTIVITIES OF LIQUIDS

Substance	Temp, C	Thermal conductivity, cgs units × 10⁶	Substance	Temp, C	Thermal conductivity, cgs units × 10⁶
n-Heptyl alcohol[a]	30	383	Bromobenzene[a]	30	306
	60	377		60	300
	100	369		100	291
n-Octane[b]	30	342	Iodobenzene[a]	30	288
	60	328		60	284
	100	309		100	280
n-Nonane[c]	30	338	Nitrobenzene[a]	30	391
	60	325		60	382
	100	312		100	365
n-Decane[a]	30	335	Ethylbenzene[a]	30	355
	60	320		60	338
	100	306		100	326
n-Decane[d]	30	351	Nitromethane[a]	30	515
	60	343		60	495
	100	328		...	...
n-Dodecane[c]	30	352	Eugenol[a]	30	384
	60	345		60	383
	100	335		100	381
Methylcyclohexane[c]	30	305	Paraldehyde[a]	30	346
	60	294		60	338
	...	...		100	322
Benzene[a]	30	380	Chlorinated diphenyl 1242[e]	30	294
	60	360		60	297
	...	...		100	300
Fluorbenzene[a]	30	343	Chlorinated diphenyl 1248[e]	30	281
	60	328		60	284
	...	...		100	288
Chlorobenzene[a]	30	346	Mobiloil *B*[f]	30	374
	60	334		60	370
	100	321		100	367

[a] Obtained from Eastman Kodak Company.
[b] Obtained from General Motors Research Laboratory.
[c] Obtained from the Pennsylvania State College.
[d] Obtained from the U. S. Bureau of Standards.
[e] Obtained from the Swann Chemical Company.
[f] Obtained from the Massachusetts Institute of Technology.

The liquids obtained from the Eastman Kodak Company were, in general, of their best purity, although some of the liquids obtained there were listed as "practical," which are considered suitable for laboratory work.

The n-decane obtained from the Bureau of Standards was given to the author by Dr. B. J. Mair, American Petroleum Institute Project No. 6, who wrote (4) that it was not of the highest purity, but was much purer than that available commercially. The thermal conductivity k of the n-decane obtained from the Eastman Kodak Company differed in value from that of the Bureau of Standards by a little less than 5 per cent. Impurities can easily account for such a difference, and the sample from the Eastman Kodak Company was of unknown purity.

The n-octane was of high purity, synthesized by Paul L. Cramer, and was supplied by T. A. Boyd of the staff of the Gen-

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[2] Numbers in parentheses refer to Bibliography at the end of the paper.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

eral Motors Research Laboratory. The thermal conductivity checks rather closely with that previously obtained by the author (1).

The two chlorinated diphenyls were given by the Swann Chemical Company. Note (Table 1) that the thermal conductivity of each has a positive temperature coefficient; only water, glycerol, and p-cymene have previously shown a positive coefficient at room temperatures.

The n-nonane, n-dodecane, and methylcyclohexane were supplied by Prof. M. R. Fenske's laboratory at the Pennsylvania State College through C. O. Tongberg. Mr. Tongberg estimated the purity of the n-nonane (n_D^{20} 1.4070) to be better than 90 mol per cent, the purity of the n-dodecane (n_D^{20} 1.4219) to be better than 97 mol per cent, and the purity of the methylcyclohexane (n_D^{20} 1.4234) to be better than 99 mol per cent.

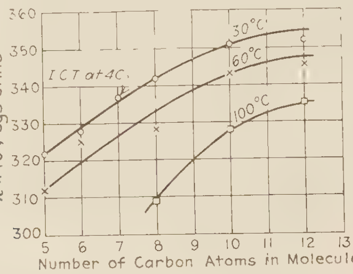

Fig. 1 Thermal Conductivity of the Paraffin Series

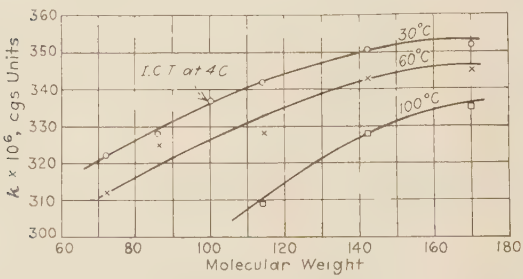

Fig. 2 Thermal Conductivity of the Paraffin Series Plotted Against Molecular Weight

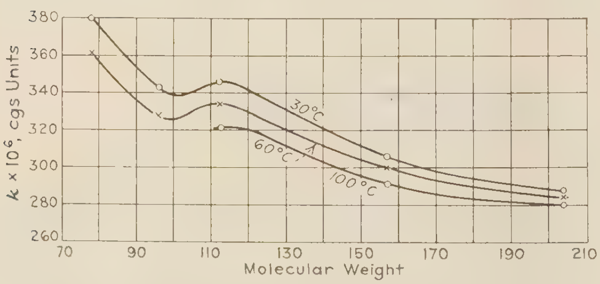

Fig. 3 Thermal Conductivity of the Benzene Series

Notice that the value for n-nonane is lower than might be expected for a pure liquid in this series. Mr. Tongberg claims (5) that the impurities expected are probably saturated cyclic compounds such as isopropylcyclohexane and 1,2,3 trimethylcyclohexane. Since methylcyclohexane has a much lower thermal conductivity than the nonane, it would be expected that the value for the 90 per cent n-nonane would be low. It is 2.5 per cent less than the value obtained from the smoothed curve, Fig. 1.

The Paraffin Series. In Fig. 1 are shown values of thermal conductivity k of the paraffin series plotted against the number of carbon atoms in the molecule. Because of the uncertainty of n-nonane, it has been excluded. Some of the values have been obtained previously: The n-hexane by Smith (1); the n-pentane by Bridgman (2); and the n-heptane from the International Critical Tables (6). A study of Fig. 1 shows that a direct correlation obviously exists. The author recommends the thermal conductivities obtained from the smoothed curves in Fig. 1 and listed in Table 2.

TABLE 2 THERMAL CONDUCTIVITY OF THE PARAFFIN SERIES

Substance	Thermal conductivity, cgs units × 10^6 At 30 C	At 60 C	At 100 C
n-pentane C_5H_{12}	322	312	...
n-hexane C_6H_{14}	329	320	...
n-heptane C_7H_{16}	336	327	...
n-octane C_8H_{18}	342	333	309
n-nonane C_9H_{20}	347	338	320
n-decane $C_{10}H_{22}$	351	343	328
n-undecane $C_{11}H_{24}$	354[a]	346[a]	333[a]
n-dodecane $C_{12}H_{26}$	355	347	335

[a] Obtained by interpolation from smooth curves in Fig. 1; no experimental verification is available to check this value.

It will be noticed that the maximum departure of the experimental points from the smoothed curves is only 1.6 per cent, and the two points which do depart by this amount lie on opposite sides of the central line.

A somewhat straighter line is obtained by plotting thermal conductivity k against molecular weight as shown in Fig. 2. Because of the 10 per cent impurity of the n-nonane, the experimental values for it have been omitted from the graphs. The 3 per cent impurity in the n-dodecane permits us to depart slightly from the observed value, since impurities are probably responsible for the slightly low value obtained (less than 1 per cent from the smooth curve). The value for n-decane is that obtained for the sample from the Bureau of Standards, since Dr. Mair (4) claims better purity for it than is available commercially.

The Benzene Series. The results of the thermal-conductivity tests of the benzene series are plotted in Fig. 3. The peculiar dip in the curves would seem to indicate an error of about 6 per cent in the determinations for fluorbenzene. This, however, is not the case, for the tests on fluorbenzene were rerun, and the values at 30 C checked to closer than 0.33 per cent; and the departure from a mean at 60 C was not much greater. As a matter of fact, benzene, fluorbenzene, chlorobenzene and bromobenzene were all run twice. Notice the gradual approach of the curves to one another as molecular weight increases. In other words, as the molecular weight increases, the temperature coefficient decreases.

Miscellaneous Information

TABLE 3 PROPERTIES OF MOBILOIL *B* AND CHLORINATED DIPHENYLS TESTED

Liquid	Molecular weight	Temp C	Density, g per cc	Viscosity, centipoises	Specific heat, cal per g per deg C
Mobiloil *B*	500	20.0	0.899	486 at 27 C	0.47
		40.0	0.887	210	0.48
		60.0	0.875	69	0.50
		80.0	0.864	30	0.52
		100.0	...	15	0.54
Chlorinated diphenyl 1242	257.35	25.0	1.38	....	..
		65.0	1.35	....	..
		37.8	...	23.60	..
		98.9	...	2.73	..
Chlorinated diphenyl 1248	292.80	15.5	1.457	....	..
		65.0	1.408	....	..
		21.1	...	414.0	..
		37.8	...	66.0	..
		54.4	...	21.5	..
		98.9	...	4.5	..

Table 3 shows the properties of Mobiloil *B* supplied by Bays (20) and information on two chlorinated diphenyls, furnished by the Swann Chemical Company (21); the values of density and viscosity in the latter case being approximate only. The properties of the other liquids tested have been published in the various references of the Bibliography.

TABLE 4 THERMAL CONDUCTIVITIES OF LIQUIDS

Liquid	Temp, C	$k \times 10^6$, cgs units	Observer[a]
n-pentane	30	322	Author
	60	312	
n-hexane	30	329	Author
	60	320	
n-heptane	30	336	Author
	60	327	
n-octane	30	342	Author
	60	333	
	100	309	
n-nonane	30	347	Author
	60	338	
	100	320	
n-decane	30	351	Author
	60	343	
	100	328	
n-undecane[b]	30	354	Author
	60	346	
	100	333	
n-dodecane	30	355	Author
	60	347	
	100	335	
Methylcyclohexane	30	305	Author
	60	294	
Gasoline No. 1	30	320	Daniloff, Smith (13)
Gasoline No. 2	30	318	Daniloff, Smith (13)
Kerosene	30	357	Bridgman (2)
	75	333	
Mobiloil *B*	30	374	Author
	60	370	
	100	367	
Rabbeth spindle oil	30	341	Smith (1)
	75	338	
	100	333	
Velocite *B*	30	341	Smith (1)
	75	338	
	100	331	
Medium cylinder oil	0	370	Kaye and Higgins (14)
	20	366	
	40	363	
	60	360	
	80	357	
	100	354	
	120	351	
	140	348	
	160	345	
	180	342	
	200	339	
Red oil	30	337	Smith (1)
	75	335	
	100	329	
Light heat-transfer oil	30	316	Smith (1)
	75	312	
	100	309	
Transformer oil	30	324	Smith (16)
	60	321	
	100	316	
Transformer oil	0	324	Kaye and Higgins (14)
	20	320	
	40	316	
	60	312	
	80	308	
	100	304	
Transformer oil No. 1	15	320	Davis (15)
	20	320	
	50	310	
	60	310	
	75	310	
Transformer oil No. 2	12	290	Davis (15)
	15	290	
	20	290	
	50	290	
	60	290	
	75	285	
Transformer oil No. 3	12	300	Davis (15)
	15	300	
	20	300	
	50	290	
	60	290	
	75	290	
Paraffin	12	300	Davis (15)
	15	300	
	20	300	
	50	280	
	60	270	
	75	260	
Paraffin oil, acid free	0	300	Kaye and Higgins (14)
	20	297	
	40	294	
	60	291	
	80	288	
	100	284	
	120	281	
Medicinal paraffin	0	301	Kaye and Higgins (14)
	20	300	
	40	300	
	60	300	
	80	299	
	100	299	
Medicinal paraffin	120	299	
	140	298	
	160	298	
	180	298	
Castor oil	0	437	Kaye and Higgins (14)
	20	432	
	40	428	
	60	424	
	80	420	
	100	415	
	120	411	
	140	406	
	160	402	
Olive oil	0	405	Kaye and Higgins (14)
	20	402	
	40	399	
	60	397	
	80	394	
	100	391	
	120	388	
	140	385	
	160	382	
	180	379	
	200	376	
Turpentine	12	303	Weber, Chree Graetz, I. C.T. (6)
Methyl alcohol	30	503	Bridgman (2)
	75	492	
Ethyl alcohol	30	433	Smith, Bridgman, Daniloff (1, 2, 3)
	75	416	
n-propylalcohol	30	409	Daniloff (3)
	75	393	
Isopropylalcohol	30	367	Bridgman (2)
	75	363	
	30	377	Smith (1)
	60	371	
n-butylalcohol	30	400	Bridgman (2)
	75	391	
Isobutylalcohol	12	375	Weber, I.C.T. (6)
n-amylalcohol	30	388	Daniloff (3)
	75	375	
	100	368	
Isoamylalcohol	30	354	Bridgman (2)
	75	348	
	30	362	Smith (1)
	75	358	
n-hexylalcohol	30	386	Daniloff (3)
	75	374	
	100	368	
n-heptylalcohol	30	389	Daniloff (3)
	75	376	
	100	372	
n-octylalcohol	30	398	Daniloff (3)
	75	387	
	100	375	
n-nonylalcohol	30	401	Daniloff (3)
	75	391	
	100	382	
Benzene	30	380	Author
	60	360	
Fluorbenzene	30	343	Author
	60	328	
Chlorobenzene	30	346	Author
	60	334	
	100	321	
Bromobenzene	30	306	Author
	60	300	
	100	291	
Iodobenzene	30	288	Author
	60	284	
	100	280	
Toluene	30	364	Bridgman (2)
	75	339	
	30	357	Smith (1)
	75	338	
o-xylene	21	373	Goldschmidt, I.C.T. (6)
m-xylene	21	370	Goldschmidt, I.C.T. (6)
Ethylbenzene	30	355	Author
	60	338	
	100	326	
Nitrobenzene	30	391	Author
	60	382	
	100	365	
Water	30	1440	Bridgman (2)
	75	1540	
	30	1440	Smith (1)
	75	1480	
	12	1400	Davis (15)
	15	1400	
	20	1400	
	40	1470	
	50	1500	
	0	1450	Kaye and Higgins (14)
	20	1490	
	40	1530	Kaye and Higgins (14)
	60	1560	
	80	1600	
	0	1320	Jakob (22)
	15	1380	
	60	1560	
	0	1390	Martin and Lang (17)
	15	1440	
	60	1590	
	8.8	1370	Schmidt and Sellschopp (24)
	18	1417	
	40	1512	
	75	1593	
	90	1625	
	105	1643	
	125	1637	
	158	1637	
	195	1603	
	243	1512	
	269	1433	
	30	1600	Bates (25)
	7.8	1347	Schmidt (27)
	78	1610	
	41.1	1495	
	72.2	1611	
	5	1372	Nukiyama and Yoshizawa (28)
	80	1575	
	8	1340	Bates (23)
	78	1620	
Carbon disulphide	30	382	Bridgman (2)
	75	362	
Carbon tetrachloride	0	263	Goldschmidt, I.C T. (6)
Chloroform	12	330	Weber, I.C T. (6)
p-cymene	12	315	Weber, I.C.T. (6)
	30	322	Smith (1)
	60	325	
Dichlormethane (methylene chloride)	0	376	Kardos (26)
	30	364	
Ethylene glycol	0	633	Goldschmidt, I.C.T. (6)
Ethyl acetate	19	415	Stankevic, I. C.T. (6)
			Weber, I.C.T. (6)
Ethyl bromide	30	286	Bridgman (2)
	75	273	
Ethyl iodide	30	265	Bridgman (2)
	75	261	
Ether	30	329	Bridgman (2)
	75	322	
Petroleum ether	30	312	Bridgman (2)
	75	302	
Glycerol, Brit. pharm.	0	673	Kaye and Higgins (14)
	20	680	
	40	687	
	60	695	
	80	702	
	100	709	
	120	716	
	140	723	
Glycerol	0	670	Kardos (8)
	20	695	Landolt-Börnstein (29)
	20	690	I.C.T. (6)
	20	680	Bates (30)
	80	680	
Methyl chloride	−15	458	Kardos (26)
	0	428	
	30	367	
Eugenol	30	384	Author
	60	383	
	100	381	
Paraldehyde	30	346	Author
	60	338	
	100	322	
Nitromethane	30	515	Author
	60	495	
Acetic acid	20	411	Lees, Weber, I.C.T. (6)
Acetone	30	429	Bridgman (2)
	75	403	
Amyl acetate	12	342	Weber, I.C.T. (6)
Aniline	0	430	Goldschmidt, Weber, Mache, and Tagger, I.C.T. (6)
	0	412	Kaye and Higgins (14)
	20	412	
	40	412	
Chlorinated diphenly 1242	30	294	Author
	60	297	
	100	300	
Chlorinated diphenyl 1248	30	281	Author
	60	284	
	100	288	

[a] The numbers in parentheses following the observer's name refers to the Bibliography at the end of the paper. [b] Obtained by interpolation from the smooth curve in Fig. 1.

Nomenclature

C_1 = inner repelling force of molecules, g
c_p = specific heat at constant pressure, cal per g per deg C
K = volume compressibility, sq cm per g
k = thermal conductivity, cal per sec per sq cm per deg C per cm
k_1 = thermal conductivity, Btu per hr per sq ft per deg F per in.
m = mass of a molecule, g
M = molecular weight
v = velocity of sound in the liquid, cm per sec
α = gas constant = 2.02×10^{-16}, ergs per deg C
ρ = density, g per cc
θ = temperature, C
μ = viscosity, centipoises
ν = kinematic viscosity, centistokes
Δ = mean distance of separation of centers of molecules, assuming an arrangement cubical on the average, cm
δ = mean distance between edges of molecules, cm

Theoretical Equations

Bridgman's Theoretical Equation. In 1923, Bridgman (2) suggested the theoretical equation

$$k = 2\alpha v/\Delta^2 \qquad [1]$$

where $\Delta = (m/\rho)^{1/3}$. And k is in ergs per sq cm per sec per deg C/cm.

Bridgman tested eleven liquids, and showed that rough values of thermal conductivity could be obtained by using Equation [1]. The average error between the calculated and observed thermal conductivities of the eleven liquids was 16.6 per cent, with a maximum error of 38 per cent.

The author has found thermal conductivities for several other liquids for which the velocity of sound was obtainable, and the check between Bridgman's theoretical values and the observed values is shown in Table 5 for 28 liquids. It will be noticed that the average error is 15.2 per cent, and the maximum error is 39 per cent. The equation, it should be noted, gives absolute cgs units; to convert to the ordinary cgs units, it is necessary only to divide by 4.183×10^7, the conversion factor for ergs to calories. Thus, it would appear that Bridgman's equation could be used satisfactorily if an accuracy of better than 39 per cent is satisfactory. It is very interesting and gratifying that an entirely theoretical equation should give results as close as those obtained to the observed values.

Debye's Theoretical Equation. In 1914, Debye (7) suggested a rather complicated equation for the conductivity of a solid which can readily be reduced to one exactly like Bridgman's by substituting the liquid values for the solid ones, as Bridgman (2) has pointed out, with the exception that the constant 2

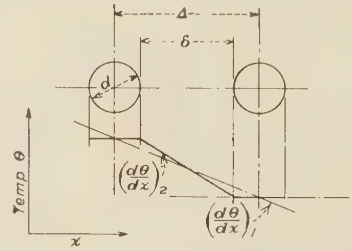

Fig. 4 Diagram of Temperature Gradients

in Equation [1] is replaced by $1/2$. Thus Debye's values are only one quarter of Bridgman's and, since the latter gives the approximate magnitude, Debye's equation was not investigated further.

Kardos' Theoretical Equation. More recently, Kardos (8) has modified Bridgman's theory, by allowing for the diameter of the molecules. In this theory the temperature gradient is increased from $(d\theta/dx)_1$, the average through the liquid, to the gradient $(d\theta/dx)_2$ as shown in Fig. 4.

Kardos' equation is

$$k = \rho c_p v \delta \qquad [2]$$

TABLE 5 CHECK OF THEORETICAL EQUATIONS

Liquid	Temp, C	$k \times 10^6$ observed	$k \times 10^6$ calculated by Bridgman's equation	$\frac{k_{calc}}{k_{obs}} \times 100$ per cent	$k \times 10^6$ calculated by Kardos' equation $\delta = 0.95 \times 10^{-8}$, cm	$\frac{k_{calc}}{k_{obs}} \times 100$ per cent	$\Delta = \left(\frac{m}{\rho}\right)^{1/3} \times 10^8$, cm	$\delta = \frac{C_1 K^{1/3}}{\Delta} \times 10^4$, cm	$k \times 10^2$	$\frac{k_{calc}}{k_{obs}} \times 100$, per cent
Water	21.5	1493	1492	100	1406	94	3.10	0.984	1456	98
Methyl alcohol	19	506	669	132	515	102	4.06	0.950	515	102
Ethyl alcohol	15	439	589	134	542	123	4.57	0.834	476	108
Propyl alcohol	20	413	449	109	504	122	4.98	0.758	402	97
n-butyl alcohol	20	402	376	94	477	119	5.32	0.739	370	92
Amyl alcohol	20	391	386	99	523	134	5.64	0.665	366	94
Carbon disulphide	20	386	522	135	334	87	4.63	0.802	282	73
Carbon tetrachloride	0	263	255	97	233	89	5.37	0.733	180	68
n-hexane	21	332	298	90	366	110	6.00	0.799	308	93
n-octane	10	348	270	78	373	107	6.42	0.672	263	76
Gasoline[a]	7.4	320	371	116	473	148	....	...	...	...
Benzene	16.3	389	408	105	393	101	5.26	0.753	312	80
Chlorobenzene	17	351	419	119	425	121	5.51	0.667	299	85
Nitrobenzene	17.5	395	479	121	603	153	5.52	0.580	369	93
Ethylbenzene	20	361	380	105	449	124	5.86	0.650	307	85
Toluene	20.5	366	409	112	435	119	5.60	0.703	322	88
Methylene chloride	0	376	455	121	360	96	....	....	...	...
Glycerine	0	673	782	116	1371	204	4.94	0.501	723	107
Ethyl ether	15	331	322	97	377	114	5.54	0.784	311	94
Ethyl bromide	30	286	348	122	271	95	5.00	0.835	238	83
Ethyl iodide	30	265	288	109	237	89	5.12	0.770	192	72
Acetone	20.5	445	469	105	471	106	4.95	0.798	396	89
Formic acid	....	648	637	98	625	95	....	...	...	...
Chloroform	15	330	368	112	317	96	5.08	0.792	264	80
Nitromethane	21	521	661	127	606	116	4.46	0.813	519	100
Aniline	20.8	412	572	139	804	195	5.32	0.582	493	120
Paraldehyde	21.5	348	318	91	500	144	6.03	0.655	340	98
Ammonia	5[b]	1200	1562	130	1276	106	....	...	...	...
Turpentine	15	303	...	...	460	152	....	...	...	...
				Average error = 15.2 per cent		Average error = 23.4 per cent				Average error = 12.5 per cent

Note: Values of k are in cgs units, cal per g per deg C.
[a] The velocity of sound was measured on another gasoline.
[b] At 12 atm pressure, abs.

This is the same as Equation [1] except that δ replaces Δ. However, the experimental values for each of the terms were introduced directly in this case into Equation [2] rather than into Equation [1]. The difficulty in checking this equation lies in the finding of suitable values for δ. Kardos suggests that, as a first approximation, δ be assumed constant, and equal to 0.95×10^{-8} cm; he makes the further statement that δ will not vary much for various liquids. Table 5 shows how calculated values for 29 liquids, whose various properties have been determined, check experimental data. The temperature is not the same in each case, and is added for convenience. The thermal conductivity of liquids varies rather slowly with change in temperature, whereas the variation in the velocity of sound with temperature is not well known. For this reason it was considered to be safer to extrapolate and interpolate thermal conductivities from reliable data to the temperature at which the velocity of sound had been determined. The extrapolation in thermal conductivity was ordinarily from 30 C, so the possible error was a small percentage of the recorded value.

The average error in this case is 23.4 per cent, and the maximum error is 104 per cent. Thus, it would appear that the assumption of constancy of δ is not tenable if reasonably accurate data were wanted. Kardos attempted to figure more accurate values for δ for alcohols. He calculated Δ from $\Delta = (m/\rho)^{1/3}$, assuming an arrangement which on the average is cubical. He also states that the inner repelling force C_1 of the molecules is unchangeable fundamentally at equal pressures and equal temperatures for allied materials, and for alcohols and water $C_1 = 8.84 \times 10^{-10}$ g. He also claimed that $\delta = C_1 K^{1/3}/\Delta$. The development of this relationship is not at all clear to the author. Kardos stated that the linear compressibility, defined as the ratio of the change in length to the original length by a change in the outside pressure of 1 kg per sq cm, is the third root of the ordinary volume compressibility K, defined as the ratio of the change in volume to the original volume by the same change in pressure. This is clearly a misstatement, for in the first place, the dimensions are wrong, and the order of magnitude is wrong. It is rather obvious that for a change of 1 kg per sq cm the change in volume divided by the original volume is $\Delta V/V$, and if $V = l^3$, then $\Delta V/V = 3l^2\Delta l/l^3 = 3(\Delta l/l)$, and the linear compressibility is one third of the volume compressibility, not the third root. In spite of this, the author worked out the thermal conductivities for several liquids using the formula Kardos suggested for alcohols and water. This extension to other liquids, of course, was just to see what would happen on this basis. Table 5 shows the results. Notice that, outside of the error in the order of magnitude of ten thousand times, the average error for the liquids is only 12.5 per cent, so that this method of calculation, although it must be classified as semiempirical, gives a better average error than the assumption of constant δ. Notice that δ calculated this way is about 3000 times Δ for water, an obvious absurdity.

It occurred to the author to investigate the possibility of using $K/3$ for the linear compressibility, using the rest of Kardos' development for the alcohols. It was found that the results obtained are again not of the proper order of magnitude, nor are the significant figures in this case anywhere near correct.

When the author noticed this, he wrote to Dr. Kardos asking for an explanation of his development of the theory, pointing out what appeared to be errors in reasoning and calculation.

Kardos reexamined his development and concluded that it must now be considered invalid. He also stated that there must be an error in C_1, that it should be 8.84×10^{-15} on the kilogram basis and not 8.84×10^{-13} as stated in the original paper (8); that the units of C_1 could no longer be considered to be kilograms, but must be $\text{kg}^{1/3} \times \text{cm}^{4/3}$. With those remarks, Kardos showed that the error in the decimal point was eliminated.

Kardos intends to continue work on this subject in an effort to obtain the physical significance of the equation; the close agreement with experiment would seem to indicate that there is such a physical meaning.

General Empirical Equations for Nonmetallic Liquids

Weber's Empirical Equation. H. F. Weber (9) suggested that

$$k = 0.00359\, c_p\, \rho \sqrt[3]{(\rho/M)} \qquad [3]$$

The values of k calculated by Equation [3] for different liquids are given in Table 6 together with experimental results. The average error between observed and calculated results is 20.1 per cent, and the maximum error is 51 per cent. Note that in only two cases is the error positive, whereas in 43 others the error is negative. For this reason the constant in the equation was changed to 0.0043, and the modified equation is

$$k = 0.0043 c_p\, \rho \sqrt[3]{(\rho/M)} \qquad [4]$$

The agreement with experiment of this modified equation is also shown in Table 6. The average error has been reduced to 14.8 per cent, and the maximum error is 41 per cent. Thus this modified equation is definitely better than Equation [3] suggested by Weber (9). It is obvious that the average error could be still further reduced by an increase in the size of the constant, but at the expense of increasing the maximum error. Thus, this modified Weber equation gives errors of the same general size as Equation [1].

Smith's Empirical Equation. In 1930 Smith (1) suggested an entirely empirical equation

$$k \text{ at } 30\text{ C} = 8.1 \times 10^{-4}\, \frac{\rho^{2.15} c_p^{1.55} M^{0.192}}{\mu^{0.12}} \qquad [5]$$

This equation satisfied all the liquids for which data were at that time available; in all, 15 liquids with reasonable variation in physical properties. The agreement with experiment for every liquid except two oils was closer than 4.5 per cent. The oils gave errors of 3 and 20 per cent, respectively. In the original paper (1), the errors of the two oils were reported as 6 and 9 per cent, respectively, these values being based on an estimated specific heat which has since been determined experimentally by Prof. L. W. Cummings of Massachusetts Institute of Technology, who supplied the author with the values obtained (10).

Now that the thermal conductivities of many more liquids have been determined, it is possible to give this equation a more rigorous test. Unfortunately, in some cases the error obtained by it is rather large, and, since the author has found that Equations [6] and [6a] of this paper give a closer check with experiment, he thinks it desirable to use them.

Smith's Dimensionless Equation. In 1931, Smith (11) derived an equation by dimensional analysis involving four dimensionless groups. At present sufficient data are not available to check it, or to find how each group affects the result.

The Author's Proposed Equation. The author could see little prospect of modifying the exponents of Equation [5] to satisfy all liquids, so he started on another track. By taking variations from approximate mean values, he found that

$$k \text{ at } 30\text{ C} = 0.000361 + \frac{(c_p - 0.45)^3}{155} + \frac{\sqrt[3]{\rho/M} - 0.20}{800} + \frac{\nu^{1/9} - 1}{10{,}000} \qquad [6]$$

This is equivalent to

$$k = 0.000011 + \frac{(c_p - 0.45)^3}{155} + \frac{\sqrt[3]{\rho/M}}{800} + \frac{\nu^{1/9}}{10{,}000} \qquad [6a]$$

TABLE 6 CHECK OF GENERAL EMPIRICAL EQUATIONS

		Weber's equation[a]		Weber's modified equation[b]		Smith's equation[c]	
Liquid	$k \times 10^6$ observed	$k \times 10^6$	$\frac{k_{calc}}{k_{obs}} \times 100$, per cent	$k \times 10^6$	$\frac{k_{calc}}{k_{obs}} \times 100$, per cent	$k \times 10^6$	$\frac{k_{calc}}{k_{obs}} \times 100$, per cent
n-pentane	322	258	80	309	96	367	114
n-hexane	329	243	74	291	88	350	106
n-heptane	336	237	71	284	85	343	102
n-octane	342	232	68	278	81	336	98
n-nonane	347	230	66	276	80	330	95
n-decane	351	228	65	273	78	327	93
n-undecane	354	225	64	270	76	323	91
n-dodecane	355	225	63	270	76	322	91
Gasoline No. 1	320	252	79	302	94		
Gasoline No. 2	318	258	81	309	97	...	..
Mobiloil *B*	374	183	49	219	59	360	96
Rabbeth spindle oil	341	192	56	230	67	325	95
Red oil	337	174	52	208	62	363	108
Transformer oil	324	235	72	281	87	361	111
Methyl alcohol	503	495	98	593	118	495	98
Ethyl alcohol	433	429	99	514	119	455	105
n-propyl alcohol	409	404	99	484	118	439	107
n-butyl alcohol	400	371	93	444	111	415	104
Isobutyl alcohol	375[d]	379	101	454	121	427	114
Isoamyl alcohol	354	340	96	407	115	400	113
Benzene	380	288	76	345	91	382	101
Chlorobenzene	346	264	76	316	91	364	105
Bromobenzene	306	268	88	321	105	310	101
Toluene	364	263	72	315	87	364	100
o-xylene	373[e]	259	69	310	83	358	96
m-xylene	370[e]	247	67	296	80	349	94
Ethylbenzene	355	253	71	303	85	353	99
Nitrobenzene	391	309	79	370	95	373	95
Water	1510	1362	90	1632	108	1659	110
Carbon disulphide	382	275	72	329	86	358	94
Carbon tetrachloride	263[f]	246	93	295	112	275	105
Chloroform	330[d]	285	86	341	112	324	98
p-cymene	322	226	70	271	84	...	...
Dichlormethane	364	318	87	381	105	373	103
Ethylene glycol	633[f]	604	95	723	114	588	93
Ethyl acetate	415[g]	317	76	380	92	373	90
Ethyl bromide	286	268	94	321	112	324	113
Ethyl iodide	265	264	100	316	119	240	91
Ether	329	291	88	349	106	370	112
Glycerol[h]	683	619	90	741	108	511	75
Paraldehyde	346	299	86	358	104	355	103
Nitromethane	515	447	87	535	104	434	84
Acetic acid	411[i]	479	117	574	140	434	106
Acetone	429	352	82	422	98	401	93
Amyl acetate	342[d]	264	77	316	92	344	101
Aniline	412	407	99	487	118	401	97
			Average error = 20.1 per cent		Average error = 14.8 per cent		Average error = 6.7 per cent

[a] See Equation [3] of the paper.
[b] See Equation [4] of the paper.
[c] See Equation [6a] of the paper.
[d] At 15 C.
[e] At 21 C.
[f] At 0 C.
[g] At 19 C.
[h] Judging from the density, the glycerol is 97 per cent pure.
[i] At 20 C.

NOTE: All values of thermal conductivity k are in cgs units, cal per g per deg C.

NOTE: Except as noted in the foregoing footnotes, the values of k are those at 30 C.

Table 6 shows how values of k calculated by using Equation [6a] agree with experimental results. A comparison of these results may be summarized as follows:

The average error is 6.7 per cent. Outside of glycerine, no liquid has an error greater than 16 per cent; 49 per cent of the liquids check experiment closer than 5 per cent; 81 per cent of the liquids check experiment closer than 10 per cent; and 98 per cent check experiment closer than 16 per cent.

For glycerol, which is not 100 per cent pure, the maximum error is 25 per cent. With 3 per cent water, this error is reduced to 22 per cent.

Equations have been proposed for specific types of liquids. Among them is the equation of the Bureau of Standards for petroleum products.

Bureau of Standards Equation for Petroleum Products

In 1929, Cragoe (12) suggested the following equation for petroleum liquids

$$k_1 = \frac{0.813}{d}\,[1 - 0.0003(t - 32)] \ldots\ldots\ldots\ldots [7]$$

where d = specific gravity of the liquid at 60 F divided by the density of water at 60 F; t = temperature, F; and k_1 = thermal conductivity, Btu per hr per sq ft per deg F per in.

Equation [7] was based on 18 petroleum oils by seven experimenters. Five of the oils were tested at the Bureau of Standards and the data, so far as the author knows, are unpublished. A few of the other values quoted by Cragoe (12) were eliminated for various reasons, but many more data have been supplied to bring the total number of petroleum liquids available for checking purposes to 26. These are listed in Table 7. The values given are for 30 C, at which temperature the equation reduces to

$$k_1 = 0.80/d \ldots\ldots\ldots\ldots [8]$$

TABLE 7 CHECK OF BUREAU OF STANDARDS EQUATION FOR PETROLEUM PRODUCTS AT 30 C

Liquid	$k_1 = \frac{0.80}{d}$	$k \times 10^6 = k_1 \times 344.2$	$k \times 10^6$ observed	$\frac{k_{calc}}{k_{obs}} \times 100$, per cent	$\frac{0.000327}{k_{obs}} \times 100$, per cent
Mobiloil *B*	0.896	308	374	82	87
Rabbeth spindle oil	0.939	323	341	95	96
Velocite *B*	0.898	309	341	91	96
Medium cylinder oil	0.895	308	365	84	90
Red oil	0.851	293	337	87	97
Light heat-transfer oil	0.879	302	316	96	103
Transformer oil (16)[a]	0.893	307	324	95	101
Transformer oil (14)	0.951	328	318	103	103
Transformer oil 1 (15)	0.962	331	318	104	103
Transformer oil 2 (15)	0.924	318	290	110	113
Transformer oil 3 (15)	0.884	304	297	102	110
Paraffin oil (14)	0.985	339	295	115	111
Medicinal paraffin (14)	0.907	312	300	104	109
Paraffin (15)	1.000	344	293	117	112
Kerosene	0.987	340	357	95	92
n-pentane	1.299	447	322	139	102
n-hexane	1.229	423	329	129	99
n-heptane	1.186	408	336	121	97
n-octane	1.153	397	342	116	96
n-nonane	1.127	388	347	112	94
n-decane	1.107	381	351	109	93
n-undecane	1.090	375	354	106	92
n-dodecane	1.052	362	355	102	92
Methylcyclohexane	1.052	362	305	119	107
Gasoline No. 1	1.146	394	320	123	102
Gasoline No. 2	1.083	373	318	117	103
				Avg error = 12.4 per cent	Avg error = 6.0 per cent

[a] Numbers in parentheses refer to Bibliography at the end of the paper.

NOTE: Values of thermal conductivity k_1 are in English units, Btu per lb per deg F, while values of k are in cgs units, cal per g per deg C.

To convert k_1 into k, that is, cgs units, multiply by 0.0003442.

Notice that the average error of the Bureau of Standards formula, Equation [8], is 12.4 per cent, with a maximum error of 39 per cent; whereas, if we assume that the value is constant and equal to 0.000327 cgs units, the average error is 6.0 per cent and the maximum error is ±13 per cent. For that reason, until more data are available, the author recommends that for petroleum products the thermal conductivity at 30 C and atmospheric pressure be assumed constant, and equal to 0.000327 cgs units.

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The Use of Alloy Steels for Side Frames and Bolsters of Freight-Car Trucks

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The author reviews briefly the use of different types of side frames for railway freight-car trucks, including arch-bar side frames, cast-steel side frames with separate journal boxes, and side frames with the journal boxes cast integral with the truss members. He discusses the present-day trend toward the use of high-tensile-strength cast alloy steel for truck side frames and bolsters, and gives results of static and dynamic tests made on truck members cast from alloy steel and A.A.R. grade-*B* steel.

THIS PAPER is a general summary of the results to date of the recent utilization of alloy steel in the manufacture of side frames and bolsters for railway freight-car trucks. The usual material of which these two truck details are made is annealed or normalized cast steel, described as grade-*B* steel in the specifications of the Association of American Railroads. As the result of independent and concerted efforts to meet high-test requirements with minimum weight for these truck parts, the designs of both have apparently reached their limit of development so that further weight reduction can be achieved only by the use of material having higher physical properties.

In order to place the results from the use of alloy steels in the proper perspective, it is necessary to trace briefly the development of truck design. The development of the cast-steel side frame dates from about 1900, and in its first more popular form was merely an attempt to substitute a single casting for the top compression member, the intermediate tension or web member and the cast struts or columns of the built-up truss of the arch-bar truck side frame shown in Fig. 1. This development, illustrated in Fig. 2, shows that the separate journal boxes used with the arch-bar side frame were retained together with the tie bar or bottom member of the truss. Structurally, if not chronologically, the next development eliminated the full-length bottom tie member and substituted short tie straps extending from under the journal boxes to brackets cast on the tension member of the side frame and riveted thereto as shown in Fig. 3.

The separate journal boxes used with arch-bar and early cast-steel trucks shown in Figs. 1, 2, and 3 were a source of annoyance and maintenance expense because of the tie bars and necessary attaching bolts. This led to the development of the jaw-type side frame shown in Fig. 4. Attempts to standardize the proportions of the jaw, the journal box, and the fastening means were not successful because of the parallel development of the integral-box side frame. This design was first proposed about 1900 but the steel-casting art had not then progressed sufficiently to make such a frame entirely satisfactory. However, with improvement in steel-casting practice it became possible to utilize this final step in side-frame simplification, and the integral-box frame is now substantially standard equipment on railway freight-car trucks.

The separate box may yet reappear when first cost is not the controlling factor. It can and usually does have a machine-finished lid face. The integral-box has a cast face, and with its use there is lost the advantage of flexibility between the frame and the box. The equalizing adjustment necessary to maintain uniform bearing area and unit pressure between journal and journal bearing must, in the integral-box side frame, be provided within the box, and this must be accomplished with no change in interior box dimensions or contained parts. The integral box also prevents any development of spring mounting for the side frames themselves and subjects these members to constant uncushioned vibration and shock.

In the course of the development of the cast-steel side frame, many experiments in the contours and cross sections of the cast truss members were made. The sections were variously of angle, T, H, or I shape and late in 1918 the U section was devised and later patented. The last-named shape is today not only typical of the designs of all manufacturers, but is prescribed in the A.A.R. specifications.

Ever since the formation of the original Master Carbuilders Association, the American railroads have progressed toward the standardization of important car details, and a few years ago the then American Railway Association sponsored a detailed design of side frame. This was later withdrawn as a definite required standard, because the art of side-frame design had not then reached its limit of development. The present requirements are merely that a freight truck shall conform to certain general dimensions and be interchangeable as a whole with any other truck meeting the same requirements.

When the then A.R.A. tentatively adopted the specific side-frame design referred to previously, there were available a number of different types of side frames, as well as hundreds of thousands of arch-bar trucks in service, representing an unknown number of variations from an earlier established standard for this particular type of truck. It was natural therefore that the A.R.A. should desire to discourage new truck designs before the chaotic conditions which existed prior to the adoption of the standard automatic coupler were duplicated in another field.

In due course it began to be appreciated that it was impossible to establish a fundamental design of a truss side frame simply by means of a static test. Side frames do not fail in service because of the slow imposition of a load several times as great as they will ever be called upon to carry. When they fail it is while carrying static loads well within their theoretical capacity under conditions of constant vibration and impact. In order to test side

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

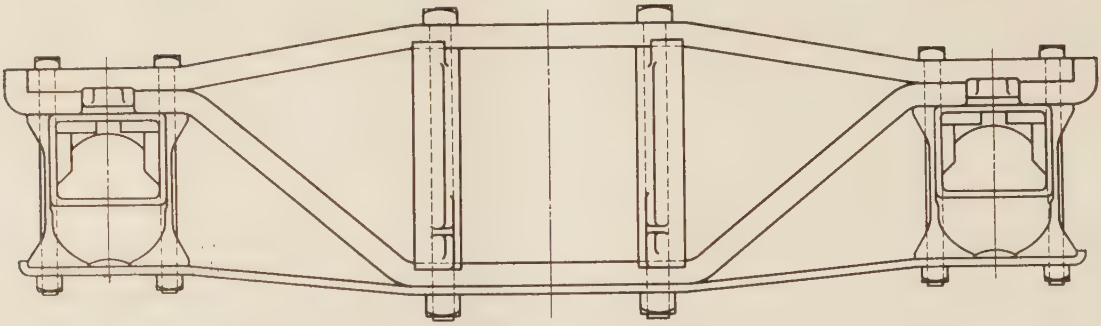

Fig. 1 Obsolete Arch-Bar Side Frame for Railway-Car Trucks

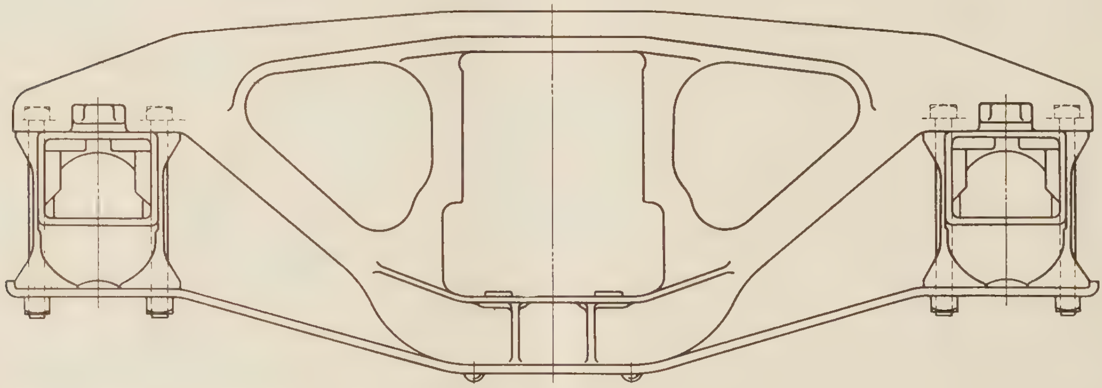

Fig. 2 Early Cast-Steel Side Frame With Separate Journal Boxes and Full-Length Bottom Tie Member

frames under conditions more nearly approximating actual service, the American Steel Foundries in 1921 built a dynamic testing machine at its plant in Granite City, Ill., and in 1923 the Symington Company designed, built, and installed at its Baltimore laboratory a similar machine but with an amplified cycle which included additional load applications.

By the use of these machines, it is possible to determine definitely on specimen frames the location and progress of the first crack, the first critical crack, and the crack leading to failure. Through long experience with the operation of these machines, and the study of test results obtained with them, tentative dynamic test specifications have been established by the A.A.R. which state that the number of loadings necessary to start a critical crack in any one specimen shall not be less than 75,000 and that the average number of loadings for four specimens shall not be less than 125,000.

Long before these specifications had reached their present form, certain conventional details of side-frame design were identified as points of weakness. As research proceeded, it was found that the establishment of a design which would give the best static and fatigue results, and be susceptible of economical manufacture through the elimination of the maximum number of foundry difficulties, required the use of a considerable number of detail patents taken out over a period of years by the various side-frame manufacturers in the course of their independent research work. Therefore, in 1927 the manufacturers, in cooperation with the then American Railway Association, cleared the path toward the development of a future side-frame standard through the formation of a manufacturers' association based on an agreement under which all detail patents on side frames and bolsters were made available to all the members as soon as any of these patents were commercially utilized. This manufacturers' association through its engineering committee develops

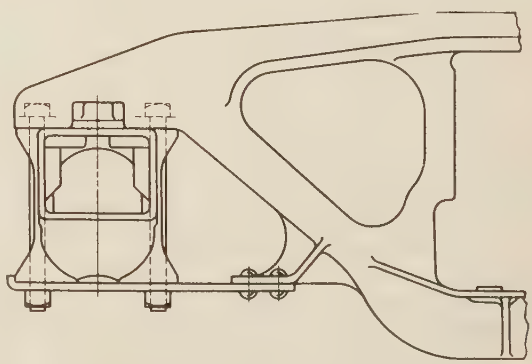

Fig. 3 A Development in Cast-Steel Side Frames With Short Tie Straps Beneath the Journal Boxes Riveted to the Tension Member

the standard designs which are available to each manufacturer-member and through continued research and cooperative test work carries on the refinement of these designs. The manufacturers' present standard design of side frame is illustrated in Fig. 5.

In 1930, at the invitation of the Car Construction Committee of the A.R.A., the members of the truck manufacturers' association began to experiment with side frames and bolsters made of steel of high tensile strength. This involved no basic change in the designs established for grade-*B* steel, but merely the reduction where possible of cross-sectional areas to suit the higher permissible unit stresses. It was foreseen that the reduction in weight could not be in inverse proportion to the increase in unit stresses for two fundamental reasons: First, the minimum thickness at any point is limited by the conditions requisite for the

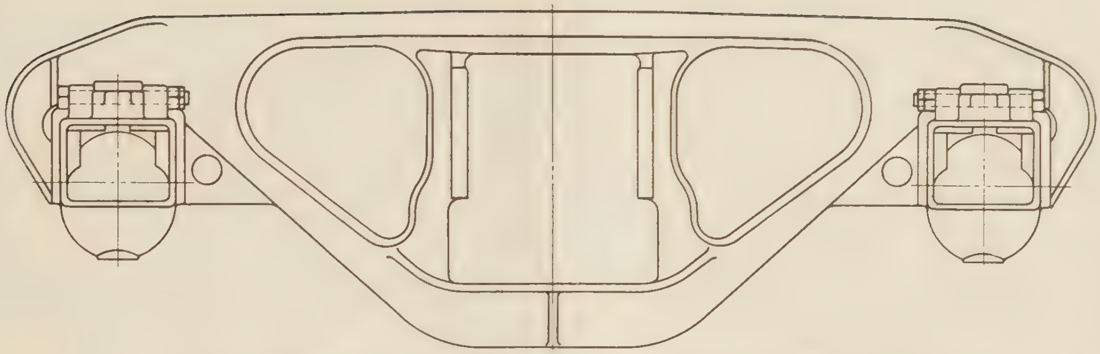

Fig. 4 Jaw-Type Cast-Steel Side Frame Designed to Eliminate Journal-Box Tie Bars and Bolts

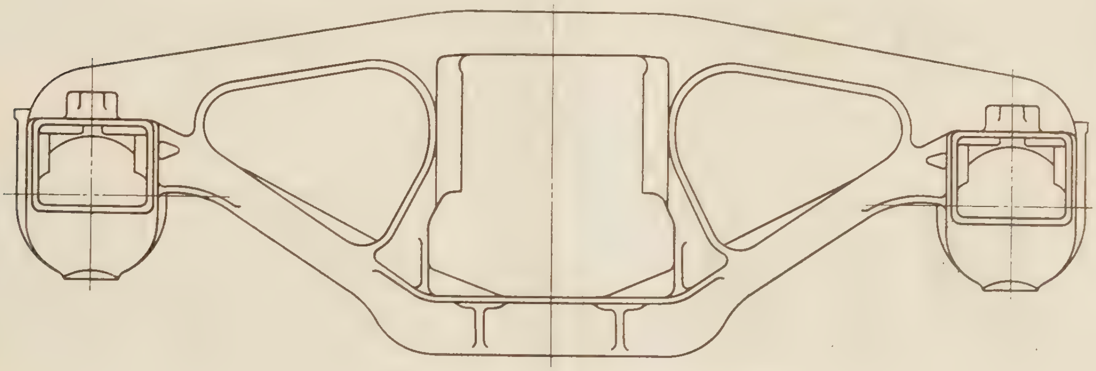

Fig. 5 The Integral-Box Cast-Steel Side Frame

production of sound castings, these conditions varying with the size of the mold, the method of gating, size and location of risers, and other details of foundry technique, as well as by the characteristics of the design itself. Second, while the side frame is theoretically a three-panel truss with bracing omitted in the middle panel to provide the standard bolster opening, the integral journal boxes, the brake-hanger brackets, and portions of the columns and spring seat are not structural members and their proportions are not determined by the truss stress calculations.

TABLE 1 RESULTS OF STATIC TESTS ON LIGHT-WEIGHT SIDE FRAMES FOR 40-TON CAR TRUCKS[a]

	Alloy steel		Grade-B steel	
	Average of four side frames	A.A.R. requirements	One side frame	A.A.R. requirements
Weight, lb	466	.....	467	.....
Transverse test:				
Deflection at 24200 lb, in.	0.091	0.120[b]	0.090	0.070[b]
Set after 38400 lb, in.	0.008	0.010[b]	0.012	0.010[b]
Vertical test:				
Deflection at 77000 lb, in.	0.039	0.060[b]	0.038	0.040[b]
Set after 144000 lb, in.	0	0.010[b]	0.008	0.010[b]
Elastic limit, lb	299700	136000[c]	194000	.....
Ultimate load, lb	484380	384000[c]	374000	384000[c]

[a] All frames cast from same pattern.
[b] Maximum.
[c] Minimum.

A general conception as to the extent to which total side-frame weight might be reduced through an increase in the unit stresses in the truss members may be had from the following analysis: For a 50-ton capacity truck, a side frame of grade-B steel weighs about 589 lb. Of this total, the two journal boxes weigh 176 lb, the two brake-hanger brackets weigh 21.5 lb, and that portion of the spring seat required for load support and distribution weighs 20.5 lb, a total of 218 lb. This leaves 371 lb for the truss members, exclusive of those portions of the journal boxes which form the necessary connections between tension and compression members. If we assume that the excluded nontruss members might, by the use of alloy steel, be reduced to a total weight of 193 lb from the original 218 lb without creating foundry difficulties or modifying certain characteristics of design essential for the cooperation of other details, and we subtract this reduced weight from the actual weight of 510 lb of a high-tensile alloy-steel frame, we have a difference of 317 lb for the alloy-steel truss members as against 371 pounds for the corresponding members of the frame of grade-B steel, which is approximately the inverse proportion of the average ultimate strengths of 98,200 and 77,730 lb per sq in., respectively, of the two materials. The ultimate strengths rather than the yield points are used in this analysis because the specifications for regular production require tests to failure. Expressed in another way, the weight of truss members of the alloy-steel side frame would be reduced 14.5 per cent while the reduction for the nontruss members would be 11.5 per cent.

At this relatively early stage in the development of high-tensile alloy-steel frames and bolsters, it is possible in only a few cases to give test figures comparing them with frames and bolsters of grade-B steel. No one manufacturer has made side frames and bolsters for the 40-, 50-, and 70-ton standard capacity trucks in both materials and from absolutely interchangeable patterns. Therefore, some of the comparisons will unavoidably be made between grade-B castings from one manufacturer and alloy-steel castings from another.

In Table 1 is given a comparison between the average results from the tests of four 40-ton alloy-steel side frames and one side frame of grade-B steel, all molded from the pattern for the alloy-steel side frame. In Table 2 are given the average results from the tests of five 50-ton frames of grade-B steel and two alloy-

TABLE 2 RESULTS OF STATIC TESTS ON NORMAL-WEIGHT SIDE FRAMES FOR 50-TON CAR TRUCKS[a]

	Alloy steel		Grade-B steel	
	Average of two side frames	A.A.R. requirements	Average of five side frames	A.A.R. requirements
Weight, lb	596	.....	592	.....
Transverse test:				
Deflection at 29000 lb, in	0.057	0.120[b]	0.060	0.070[b]
Set after 48000 lb, in	0.005	0.010[b]	0.006	0.010[b]
Vertical test:				
Deflection at 95000 lb, in	0.034	0.060[b]	0.034	0.040[b]
Set after 180000 lb, in	0	0.010[b]	0.001	0.010[b]
Elastic limit, lb	371000	170000[c]	245400	.....
Ultimate load, lb	762500	480000[c]	562800	480000[c]

[a] All frames cast from same pattern.
[b] Maximum.
[c] Minimum.

TABLE 3 PHYSICAL PROPERTIES OF ALLOY AND GRADE-B STEEL FOR CAR TRUCKS

	Alloy steel		Grade-B steel	
	Average of 42 heats	A.A.R. requirements	Average of 48 heats	A.A.R. requirements
Tensile strength, lb per sq in	98200	90000	77730	70000
Yield point, lb per sq in	65340	60000	43690	38000
Elongation in 2 in., per cent	26.2	22	29.0	24
Reduction of area, per cent	55.2	45	49.2	36

TABLE 4 COMPARATIVE TEST PERFORMANCE OF 40-TON-CAPACITY SIDE FRAMES AND BOLSTERS OF ALLOY AND GRADE-B STEEL

	Alloy steel		Grade-B steel	
	Average of four	A.A.R. requirements	Average of four	A.A.R. requirements
Side-frame static tests:				
Weight, lb	466	.....	529	.....
Transverse test:				
Deflection at 24200 lb, in	0.091	0.120[a]	0.068	0.070[a]
Set after 38400 lb, in	0.008	0.010[a]	0.006	0.010[a]
Vertical test:				
Deflection at 77000 lb, in	0.039	0.060[a]	0.029	0.040[a]
Set after 144000 lb, in	0	0.010[a]	0.001	0.010[a]
Elastic limit, lb	299700	136000[b]	226750	.....
Ultimate load, lb	484380	384000[b]	533950	384000[b]
	Average of two		One test	
Side-frame dynamic tests:				
Weight, lb	466	.....	516	.....
Loadings for first critical crack	209500	125000[b]	160920	100000[b]
	Average of four		Average of three	
Bolster static tests:				
Weight, lb	563	.....	690	.....
Transverse test:				
Deflection at 67000 lb, in	0.078	0.120[a]	0.051	0.075[a]
Set after 124000 lb, in	0.004	0.025[a]	0.002	0.025[a]
Vertical test at side bearing:				
Deflection at 98000 lb, in	0.051	0.080[a]	0.041	0.055[a]
Set after 155000 lb, in	0.003	0.025[a]	0.004	0.025[a]
Vertical test at center plate:				
Deflection at 98000 lb, in	0.085	0.120[a]	0.064	0.075[a]
Set after 186000 lb, in	0.003	0.025[a]	0.004	0.025[a]
Ultimate load, lb	403875	341000[b]	418666	341000[b]

[a] Maximum.
[b] Minimum.

TABLE 5 COMPARATIVE TEST PERFORMANCE OF 50-TON-CAPACITY SIDE FRAMES AND BOLSTERS OF ALLOY AND GRADE-B STEEL

	Alloy steel		Grade-B steel	
	Average of two	A.A.R. requirements	Average of five	A.A.R. requirements
Side-frame static tests:				
Weight, lb	512	.....	592	.....
Transverse test:				
Deflection at 29000 lb, in	0.075	0.120[a]	0.060	0.070[a]
Set after 48000 lb, in	0.007	0.010[a]	0.006	0.010[a]
Vertical test:				
Deflection at 95000 lb, in	0.048	0.060[a]	0.034	0.040[a]
Set after 180000 lb, in	0.002	0.010[a]	0.001	0.010[a]
Elastic limit, lb	313500	170000[b]	245400	.....
Ultimate load, lb	555900	480000[b]	562800	480000[b]
	Average of two		Average of two	
Side-frame dynamic tests:				
Weight, lb	501	.....	594	.....
Loadings for first critical crack	159642	125000[b]	277900	100000[b]
	Average of two		Average of two	
Bolster static tests:				
Weight, lb	601	.....	802	.....
Transverse test:				
Deflection at 82000 lb, in	0.092	0.120[a]	0.056	0.075[a]
Set after 154000 lb, in	0.014	0.025[a]	0.005	0.025[a]
Vertical test at side bearing:				
Deflection at 120500 lb, in	0.055	0.080[a]	0.045	0.055[a]
Set after 192500 lb, in	0.006	0.025[a]	0.009	0.025[a]
Vertical test at center plate:				
Deflection at 120500 lb, in	0.096	0.120[a]	0.069	0.075[a]
Set after 231000 lb, in	0.006	0.025[a]	0.007	0.025[a]
Ultimate load, lb	440550	423500[b]	540500	423500[b]

[a] Maximum
[b] Minimum

TABLE 6 COMPARATIVE TEST PERFORMANCE OF 70-TON-CAPACITY SIDE FRAMES AND BOLSTERS OF ALLOY AND GRADE-B STEEL

	Alloy steel		Grade-B steel	
	Average of two	A.A.R. requirements	Average of two	A.A.R. requirements
Side-frame static tests:				
Weight, lb	604	.....	711	.....
Transverse test:				
Deflection at 35000 lb, in	0.092	0.120[a]	0.063	0.070[a]
Set after 60000 lb, in	0.004	0.010[a]	0.005	0.010[a]
Vertical test:				
Deflection at 117500 lb, in	0.049	0.060[a]	0.037	0.040[a]
Set after 225000 lb, in	0.002	0.010[a]	0.002	0.010[a]
Elastic limit, lb	356000	212500[b]	260500	.....
Ultimate load, lb	657750	600000[b]	683500	600000[b]
	Average of two		Average of two	
Side-frame dynamic tests:				
Weight, lb	607	.....	698	.....
Loadings for first critical crack	227280	125000[b]	255200	100000[b]
	Average of two		Average of three	
Bolster static tests:				
Weight, lb	774	.....	959	.....
Transverse test:				
Deflection at 101000 lb, in	0.081	0.120[a]	0.053	0.075[a]
Set after 192000 lb, in	0.003	0.025[a]	0.003	0.025[a]
Vertical test at side bearing:				
Deflection at 149000 lb, in	0.058	0.080[a]	0.047	0.055[a]
Set after 240000 lb, in	0.008	0.025[a]	0.022	0.025[a]
Vertical test at center plate:				
Deflection at 149000 lb, in	0.091	0.120[a]	0.072	0.075[a]
Set after 288000 lb, in	0.008	0.025[a]	0.007	0.025[a]
Ultimate load, lb	631150	528000[b]	626000	528000[b]

[a] Maximum.
[b] Minimum.

steel frames, all molded from the pattern for grade-*B* steel frames. The figures in Tables 1 and 2 are interesting inasmuch as they show the relation between the physical properties of frames made from the two materials, which relation might be expected because of the different physical properties of tensile-test specimens made of the two materials. Physical properties of tensile-test specimens of both materials, as well as the A.A.R. minimum requirements, are given in Table 3.

Tables 4, 5, and 6 give the A.A.R. test requirements for side frames and bolsters of grade-*B* and alloy steel, and the average results from tests of actual side-frame specimens. As previously stated, these comparisons will change slightly after it has been possible to test a larger number of specimens. The number of specimens from which the figures are averaged is given in each column so that due allowance may be made for the narrow range of the tests so far reported. It is felt however that any conclusions reached from a comparison of these figures will be reasonably sound and provide a basis for determining the commercial justification for the reduction in truck deadweight through the use of alloy steel.

Much of the data herein was supplied by the American Steel Foundries, The Buckeye Steel Castings Company, and the Gould Coupler Corporation, to whom grateful acknowledgment is made.

The Interpretation of Creep Tests for Machine Design

By C. RICHARD SODERBERG,[1] PHILADELPHIA, PA.

The author presents a method of interpreting creep-test data and applies the method to several problems of machine design. The method involves a rational theory of plastic flow in polycrystalline materials to which empirical results from actual tests can be applied. The basic premise of the theory proposed is that the facts already established for plastic flow at "normal temperature" remain valid for higher temperatures as well.

Introduction

THE PROBLEM of strength at elevated temperatures has now been under active consideration for many years, not only in the steam industry but in other industries as well. A great deal of money has been expended for tests of various descriptions, and an extensive amount of experimental material is ready for study. The efforts on the part of the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on the Properties of Metals to make at least some of this material available to a wider circle of engineers is a most laudable one, and it is certain to yield constructive results.

In the following is presented a method of interpreting creep tests against a background of some of the problems most commonly encountered in the steam industry. It is freely admitted that certain aspects of the theory of plastic flow proposed cannot be fully appraised until more tests have been made, and some of the conclusions drawn with regard to the characteristics of the typical results of long-time creep tests may possibly be contradicted by results not available to the author, or by tests which will be made in the future. The designer of power machinery is forced to make the best use possible, however, of the material at hand, and the progress of the industry does not always wait for satisfactory theories.

One of the most important objects of this presentation, besides the pressing need for reasonable methods in machine design, is to encourage an appreciation for the need of rationally planned creep tests in the future. At the present time there is a woeful lack of test series in which the influence of the many variables involved is clearly brought out, and where use has been made of test specimens with a known, common thermal history prior to the test. The most serious shortcoming is the lack of information on the influence of temperature upon the different functions and constants involved. As our knowledge of the nature of the phenomena involved is increased, there should be less excuse than in the past for haphazardly planned test programs.

There is a considerable amount of literature on the subject of recent years which is worthy of attention. Perhaps the most outstanding is the recent contribution by Bailey.[2] Attention should also be called to the various papers by Nádai,[3] Odquist,[4] MacCullough,[5] and others. The rudiments of the proposed theory were presented in a discussion submitted on the paper by Bailey.[2]

The interpretation given in the following differs in certain essential respects from that given previously by the author.[6] While some of the important conclusions still remain valid, the realization that strain hardening does not appear to play the important rôle assigned to it has altered rather fundamentally the interpretation of the test results. The methods proposed in these papers will remain valid to the extent that strain hardening can be proved to influence the creep, but a closer examination of certain features of the test results has suggested that this influence is smaller than anticipated previously. Common intellectual honesty makes this acknowledgment appropriate.

Nomenclature

Any consistent set of units may be used in the application of the results given in the paper. While the choice of units may be arbitrary, the following have received common sanction in the American literature: Stress in units of pounds per square inch, strain in nondimensional units; time designated as h; and strain rate designated as $1/h$.

Following is a list of the most essential of the symbols used. Small finite increments of some of the variables are denoted by $\Delta\epsilon$, $\Delta\sigma$, etc.

---

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Contributed jointly by the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on Properties of Metals and the Applied Mechanics Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

[2] "The Utilization of Creep Test Data in Engineering Design," by R. W. Bailey, preprinted and read before The Institution of Mechanical Engineers, November, 1935. Abridged in *Engineering*, vol. 140, 1935, pp. 595 and 647; and in *The Engineer*, vol. 160, July to December, 1935, p. 592. Also, "Design Aspect of Creep," by R. W. Bailey, *Journal of Applied Mechanics*, vol. 3, March, 1936, Trans. A.S.M.E., vol. 58, p. A-1.

[3] "The Creep of Metals—I," by A. Nádai, Trans. A.S.M.E., vol. 55, 1933, paper APM-55-10, p. 61. Also, "The Creep of Metals—II," by A. Nádai and E. A. Davis, *Journal of Applied Mechanics*, vol. 3, March, 1936, Trans. A.S.M.E., vol. 58, p. A-7.

[4] "Plasticitetsteori Med Tillämpningar," by F. Odquist, monograph published by Ingeniörsvetenskapsakademien (IVA), Stockholm, 1934.

[5] "Applications of Creep Tests," by G. H. MacCullough, Trans. A.S.M.E., vol. 55, 1933, paper APM-55-12, p. 87.

[6] "Working Stresses," by C. R. Soderberg, Trans. A.S.M.E., vol. 55, 1933, paper APM-55-16, p. 131. Also, "Zulässige Benspruchungen im Maschinenbau," by C. R. Soderberg, *Schweizerische Bauzeitung*, vol. 104, 1934, p. 127.

A_1, A_2 = designations for $\partial s/\partial\sigma_1$ and $\partial s/\partial\sigma_2$
a_{11}, a_{12}, a_{22} = designations for $\partial A_1/\partial\sigma_1$, etc.
a, b = inner and outer radius, respectively, of thick ring
b_{11}, b_{12}, b_{22} = designations for $A_1^2 \dfrac{\partial S}{\partial s} + a_{11}S$, etc.
c = a factor, giving the ratio between shear strain and shear stress = a constant
E = modulus of elasticity
e = the strain invariant $= \sqrt{3}/2\sqrt{(\epsilon_1 - \epsilon_2)^2 + \ldots}$ The base of natural logarithms. The use of these quantities is such that confusion will not occur
$G(r), G(\xi)$ = functions of r and ξ
$k^{\xi}_{r} = G(r)/G(\xi)$ = a function of r and ξ which is unity when $r = \xi$
M, N = designations for $E(b_{11} + \nu b_{12})$ and $E(b_{12} + \nu b_{22})$
P, Q = functions of r and ξ associated with differential plastic flow of a thick ring
p, q = radial stresses or internal pressure
r, ξ = variable radii of cylinder problems. Mean radius of thin tube = r
S = a dimensionless function of the stress
s = the stress invariant $s = 1/\sqrt{2}\sqrt{(\sigma_1 - \sigma_2)^2 + \ldots}$
s_1 = a material constant stress
T = a dimensionless function of time
t = time
v = strain rate
δ = thickness of thin tube
ϵ = plastic strain. Principal strains = ϵ_1, ϵ_2, and ϵ_3
ν = Poisson's ratio, $\nu = 0.3$
ζ = plastic radial deformation of a thick ring; ζ' and $\dot{\zeta}$ denote differentiation with regard to T and t, respectively
σ = stress. Principal stresses = σ_1, σ_2, and σ_3. Specially designated quantities are written as σ^* with various indexes. Initial stress = σ_0

Theory of Plastic Deformation

It is in the nature of the problem that direct tests of general three-dimensional cases of stress applications cannot be made. Moreover, until a reasonable theory can be established for the general case, the data gathered from the different tests that can be made will have a somewhat restricted application. One of the most important phases of the problem to be established, therefore, is a rational theory of plastic flow in polycrystalline materials, to which the empirical results from the actual tests made can be applied.

The basic premise of the theory proposed is that as far as plastic flow is concerned the phenomena undergo no fundamental change when the temperature is increased beyond the range defined as "normal temperatures." This statement is not to be taken in the sense that the functional relationship between the different variables remains exactly the same. The implication is rather that as the temperature is increased beyond the normal range, the phenomenon of plastic flow, which at normal temperature is imperceptible for low stresses, is of dominating significance at all stresses.

At the present time it is impossible to assert that this basic premise rests on firm experimental proofs. It appears to represent a reasonable starting point, however, until its truth or untruth has been demonstrated experimentally.

On this basis, the laws of stationary plastic flow in crystalline isotropic materials will remain as already formulated.[7] These may be stated as follows:

1 The directions of the principal extensions coincide with those of the principal stresses.

2 The density or volume of the material remains constant.

3 The three principal shear strains are proportional to the three principal shear stresses.

4 The yielding follows the von Mises-Hencky criterion in which the strain energy, or the octahedral shear stress, is the determining variable.

The first three laws may be given the following mathematical interpretation

$$\frac{\epsilon_1 - \epsilon_2}{\sigma_1 - \sigma_2} = \frac{\epsilon_2 - \epsilon_3}{\sigma_2 - \sigma_3} = \frac{\epsilon_3 - \epsilon_1}{\sigma_3 - \sigma_1} = c \ldots\ldots\ldots [1]$$

$$\epsilon_1 + \epsilon_2 + \epsilon_3 = 0 \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

Out of these relations it is possible to develop the following expressions for the three principal strains

$$\left.\begin{aligned} \epsilon_1 &= \frac{c}{3}(\sigma_1 - \sigma_2 + \sigma_1 - \sigma_3) \\ \epsilon_2 &= \frac{c}{3}(\sigma_2 - \sigma_1 + \sigma_2 - \sigma_3) \\ \epsilon_3 &= \frac{c}{3}(\sigma_3 - \sigma_1 + \sigma_3 - \sigma_2) \end{aligned}\right\} \ldots\ldots\ldots [3]$$

The fourth law is yet to be formulated mathematically. The essential feature of the proposed theory is to regard the fundamental creep relation between stress and strain as applying to the following invariants

$$s = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \ldots\ldots [4]$$

$$e = \frac{\sqrt{2}}{3}\sqrt{(\epsilon_1 - \epsilon_2)^2 + (\epsilon_2 - \epsilon_3)^2 + (\epsilon_3 - \epsilon_1)^2} \ldots\ldots [5]$$

which are proportional to the shearing stress and shearing strain, respectively, in the octahedral plane.

Introducing Equation [1] into Equation [5], it is found that the fourth law of plastic flow is represented by

$$e = \frac{2}{3}cs \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

In other words, the factor of proportionality c is also proportional to the ratio of octahedral shearing strain to octahedral shearing stress. This gives a value of c for each point of the body. This value may now be introduced into Equations [3]. Carrying out the transformation and remembering that

$$\left.\begin{aligned} \frac{\partial s}{\partial\sigma_1} &= \frac{\sigma_1 - \sigma_2 + \sigma_1 - \sigma_3}{2s} \\ \frac{\partial s}{\partial\sigma_2} &= \frac{\sigma_2 - \sigma_1 + \sigma_2 - \sigma_3}{2s} \\ \frac{\partial s}{\partial\sigma_3} &= \frac{\sigma_3 - \sigma_1 + \sigma_3 - \sigma_2}{2s} \end{aligned}\right\} \ldots\ldots\ldots [7]$$

[7] "Plasticity," by A. Nádai, McGraw-Hill Book Company, New York, N. Y., 1931, Chapter 14. Also, "Theories of Strength," by A. Nádai, Trans. A.S.M.E., vol. 55, 1933, paper APM-55-15, p. 111.

it is found that these laws of plastic flow are embodied in the following simple relations

$$\left.\begin{aligned}\epsilon_1 &= \frac{\partial s}{\partial \sigma_1} e \\ \epsilon_2 &= \frac{\partial s}{\partial \sigma_2} e \\ \epsilon_3 &= \frac{\partial s}{\partial \sigma_3} e\end{aligned}\right\} \qquad [8]$$

The experimental data which serve as a basis of application of this theory are the results of long-time creep tests, which for the one-dimensional case give ϵ_1 as a function of time for different values of σ_1. In this case, $\sigma_2 = \sigma_3 = 0$, $\sigma_1 = s$, $\partial s/\partial\sigma_1 = 1$, $\partial s/\partial\sigma_2 = \partial s/\partial\sigma_3 = -1/2$, $\epsilon_2 = \epsilon_3 = -1/2\,\epsilon_1$, $\epsilon_1 = e$. Therefore, the functional relationship between e and s is identical with the experimental relationship between strain, time, and stress, which is obtained from the ordinary long-time creep test.

CHARACTERISTICS OF THE INFORMATION AVAILABLE FOR LONG-TIME CREEP TESTS

Before a practical application can be made of the foregoing theory, it is necessary to enter into the characteristic features of the results available from long-time creep tests.

This is the field in which the major efforts have been made by those interested in the problem, and a large amount of material is available for study. Unfortunately, only a small portion of this material has been accumulated against a background of rational research. Many of the tests were made with the purpose of obtaining direct comparisons between different materials from a very limited point of view, so that no attempts were made to bring out clearly the influence of stress, strain, temperature, and time, not to mention such factors as the thermal history of the material prior to the test. These data, nevertheless, are all that we have to work with, and it is necessary to make the best of this situation until more data are available.

It is not within the scope of this paper to give a comprehensive review of these data, although such a review would be most valuable. Fig. 1 gives a typical set of results of long-time creep tests. These curves were obtained a few years ago by McVetty and apply to 12 per cent Cr low-carbon steel at 850 F. The test points are not shown on the curves, but the tests were sufficiently consistent to leave little doubt of the configuration of the curves. Many of the tests reported in the literature show a scattering so great as to exclude rational interpretation. In some instances, this scattering is undoubtedly due to poor methods of tests, while in others they must be attributed to erratic behavior of the material itself. It goes without saying that such examples must be left out of the reckoning until the reason for the scattering has been found. There are enough long-time creep tests, however, which give smooth consistent curves, to justify the assertion that for most materials there is a functional relationship between the variables involved, namely, stress, strain, and time. Some conclusions must be made with regard to these relations before the results can be applied to practical problems.

All long-time creep test results have one feature in common in that the plastic deformation proceeds at a very rapid rate at first, approaching a more permanent rate of deformation at a later stage. The relative stability of the creep rate after the initial period of rapidly varying rate has served as a justification for extrapolating to values of time far beyond the period covered by the test. The existence of an inflection point, after which the creep rate will increase, makes such extrapolation somewhat dangerous, but until more information is available the designer has no other choice but to make the most reasonable extrapolation possible.

There has also been a general tendency to ignore the initial period of varying creep rate and to treat the problem as one of constant rate. It is evident that this procedure is unsatisfactory for all cases where the stresses are influenced by the plastic deformations themselves. This is nearly always the case in practical problems.

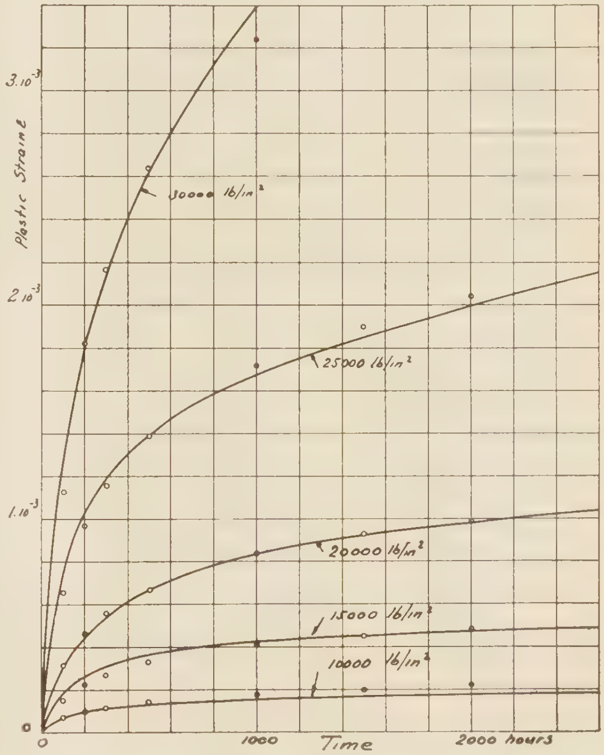

FIG. 1 CREEP CURVES FOR 12 PER CENT Cr STEEL AT 850 F

It is considered essential, therefore, to find some reasonable explanation for the rapid decrease in the creep rate with time, which occurs in the initial stage of the deformation.

The most obvious explanation that comes to mind in this connection is the phenomenon of strain hardening. The writer has personally advanced this explanation and obtained results useful for design work on this basis. This explanation implies the assumption that if the strain hardening problem were absent, the creep curves would be essentially straight lines, the slope being a function of the stress. The curvature of the strain-time relation for constant stress would then be an evidence of the influence of the strain hardening. As long as the attempt to fit creep curves to some sort of empirical function is not driven beyond a certain point, this theory does not run into direct contradiction, particularly if the experimental points are somewhat scattered.

However, a closer examination has revealed that this explanation fails to reproduce one common characteristic of all creep tests. It is evident that on the assumption of strain hardening just mentioned, the curves for high stresses and large deformations should bend more rapidly than those for low stresses. This is definitely not the case; the curves for low stress show an equally decided curvature as those for high stress. Fig. 1 shows this tendency very clearly, and in spite of the erratic nature of many creep tests, most results agree on this point. This fact appears to rule out strain hardening as an important factor.

The simplest interpretation that can be made is that all of the creep curves for any one group of specimens with a common thermal history are geometrically similar. This simple interpretation checks quite satisfactorily all the reliable data to which the author has had access.

The mathematical expression for this characteristic of the results of long-time creep tests is that the plastic elongation may be written

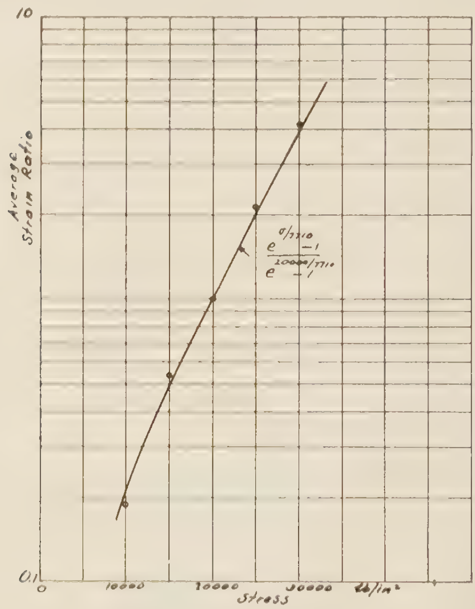

FIG. 2 AVERAGE STRAIN RATIOS FOR 12 PER CENT Cr STEEL AT 850 F

$$\epsilon = ST \ldots\ldots\ldots\ldots\ldots\ldots\ldots [9]$$

where S is a function of stress only, and T is a function of time only.

It is not the intention at the present time to read into Equation [9] a very definite physical significance. One cannot escape the conclusion, however, that the relative accuracy by which creep curves can be fitted by this interpretation reveals a fact of significance. The material undergoes structural changes under the combined influence of time and temperature, so that at the end of an extended creep test we no longer deal with the same material. This conclusion, together with the conclusion that the thermal history of the material prior to the test is of fundamental significance, must at any rate be constantly remembered in the interpretation of creep data. It offers an explanation why the results of materials with identical chemical composition so often differ very fundamentally.

Under the interpretation suggested by Equation [9], the change of structure would be practically independent of stress and strain. This conclusion is supported by tests up to certain limiting values of stress and strain. It is almost certain, however, that both stress and strain have some, although less pronounced, influence upon the change in structure. It is not unlikely that the influence of strain hardening becomes pronounced at the lower temperatures. This question must be left in abeyance, however, until further research has been made with this point of view in mind.

For most of the creep tests of interest to machine designers, the early part of the structural change is equivalent to hardening, that is, the creep curves tend to flatten out. This tendency is probably always reversed into an annealing influence after the passage of a certain time at the temperature in question.

The empirical functions S and T are easily constructed. If in Fig. 1 the curve for $\sigma = 20{,}000$ lb per sq in. is selected as a basis, a set of average ratios is obtained which, under the above interpretation, should give a consistent function of the stress. Fig. 2 represents a semilogarithmic plot of this function. It reaffirms the well established fact that arithmetical increments of stress produce geometrical increments of strain. In order to obtain a stress function which becomes zero for zero stress, however, it is necessary to modify the simple logarithmic relation somewhat. On the basis of the data examined, it is proposed to write the stress function in the following form

$$S = \frac{s_1}{E}\left(e^{\sigma/s_1} - 1\right) \ldots\ldots\ldots\ldots [10]$$

so that

$$\epsilon = \frac{s_1}{E}\left(e^{\sigma/s_1} - 1\right) T \ldots\ldots\ldots\ldots [11]$$

where s_1 is an empirical constant having the dimension of a stress.

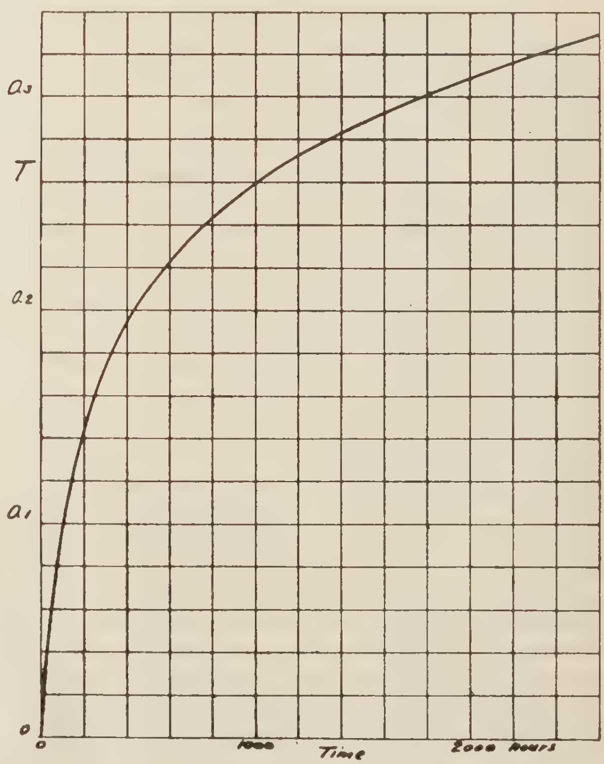

FIG. 3 TIME FUNCTION T FOR 12 PER CENT Cr STEEL AT 850 F

For the material covered by Fig. 1, this constant has the value $s_1 = 7710$ lb per sq in., as obtained from Fig. 2.

A power function of the form $S = A\sigma^n$ is regarded by many authorities as a more satisfactory form for the stress function S. When the available test data indicate that such a function fits the experimental points more consistently, there are no practical reasons why it cannot be used. The general results obtained in the following may be equally well applied to a power function as to Equation [10]. When the creep data are treated in the manner proposed here, however, it is usually not possible to obtain a satisfactory fit with a power function.

The time function T is now obtained directly from any one of the test curves. It is a dimensionless quantity, shown for the material in question by Fig. 3. The points marked by small circles in Fig. 1 have been computed from Equation [11] using $E = 30 \times 10^6$ lb per sq in., $s_1 = 7710$ lb per sq in., and taking values of T from Fig. 3. While the agreement is not perfect, it is close enough to warrant the conclusion that Equation [11] contains the essential elements of the creep phenomenon.

This analysis has been made for a large number of creep tests. In some instances the curve fit is nearly perfect, while in other instances it is of the character shown in Fig. 1. In no case of reasonably smooth test curves have there been serious discrepancies, except for very large stresses, where the observed plastic deformations usually are greater than those given by Equation [11]. This is usually within the range where the creep curves have an inflection point, and the discrepancy is undoubtedly associated with the condition of instability indicated by this inflection point.

It is natural to speculate upon the probable equation for the time function T. It has been proposed by McVetty[8] to write the equation for the creep curves in the form $\epsilon = \epsilon_0 + v_0 t$, where ϵ_0 and v_0 are functions of the stress. A similar form is suggested in a companion paper presented by Weaver.[9] Until more information is available, however, it would appear wise not to insist upon a mathematical form for this function. As will appear in the sequel, the process of direct integration of the fundamental creep relations in Equations [8] is beyond immediate hope, so that step-by-step calculations are necessary. The experimental form of T is then equally suitable as a basis for the calculation as an empirical equation.

Many of the creep tests have been interpreted against the minimum strain rate recorded in the test. When the creep curves may be replaced by straight lines, they may be expressed by the simpler relation

$$\frac{d\epsilon}{dt} = \frac{s_1}{E}\left(e^{\sigma/s_1} - 1\right)v_1 \quad\quad [12]$$

where v_1 is another material constant having the dimension of a strain rate $(1/h)$. This constant is independent of the stress, which is merely another way of saying that the actual strain rate is an exponential function of the stress. It is a restatement of the frequent observation that arithmetical increments in stress correspond to geometrical increments in the strain rate.

An important conclusion from the foregoing interpretation of creep tests is that if the specimen were subjected to the operating temperatures without stress for some appreciable time, the initial stages of the creep should be much abbreviated. No such tests have so far been made, as far as the author is aware. This type of tests offers a rich field for gaining additional knowledge of the relations involved.

Before leaving this subject, it may be well to point out that s_1 and T (or v_1) are profoundly influenced by the temperature. The nature of the influence of the temperature is really one of the most important objects of further creep tests.

Practical Application of the Results

Constant Stress in One Dimension. The case of constant one-dimensional stress application represents the basis for all other cases. In design work it is a matter of direct application of the available creep tests. It is usually necessary to extrapolate the curves to values of the time much greater than those of the test. McVetty[8] has already proposed a method for this process, which appears satisfactory, provided that our data in the future become fortified by tests over very long periods of time. The same extrapolation in time is, of course, necessary for cases of variable stress as well.

Relaxation for One-Dimensional Case. This is a very important case which deserves more attention in the future than it has received in the past. The typical application of this problem is the bolting problem. Consider a bolt having a certain initial stress. What will be the stress after the passage of a certain time?

TABLE 1

RELAXATION of BOLT							Page 1	
Material: 12% CR STEEL AT 850°F			$\sigma = 50000$	$s_1 = 7710$		$E = 30 \cdot 10^6$	$\frac{\Delta\sigma}{\Delta T} = -s_1 \frac{e^{\sigma/s_1}-1}{1+Te^{\sigma/s_1}}$	
$\Delta T\, 10^3$	$T\, 10^3$	t hours	σ lb/in²	$\frac{\sigma}{s_1}$	e^{σ/s_1}	Te^{σ/s_1}	$\frac{\Delta\sigma}{\Delta T} \cdot 10^6$	$\Delta\sigma$ lb/in²
0.40	0	0	50000	6 5	665	0	5 11	2000
0.61	0 4	—	48000	6 23	509	0 204	3 27	2000
0 93	1.01	—	46000	5 97	390	0 394	2 15	2000
1 37	1.94	—	44000	5.71	301	0 583	1 46	2000
2 00	3.31	—	42000	5 44	230	0 761	1.00	2000
2 84	5 31	—	40000	5.19	179.6	0 952	0 705	2000
4.02	8.15	—	38000	4 93	138 5	1 130	0 497	2000
5 64	12 17	—	36000	4 67	106.8	1 300	0 354	2000
7.66	17 81	10	34000	4 41	82 2	1 461	0 261	2000
10.1	25 47	15	32000	4 15	63 4	1 680	0 179	2000
14.8	35.6	20	30000	3 89	49 0	1.740	0 135	2000
20.5	50.4	35	28000	3.63	37.7	1.900	0 0975	2000
28.2	70.9	60	26000	3.37	29 1	2 055	0 0710	2000
39 0	99.1	100	24000	3 11	22 4	2 220	0 0513	2000
54.0	138 1	180	22000	2.85	17 3	2 390	0 0371	2000
37 6	192 1	380	20000	2 59	13 35	2 570	0 0267	1000
44.6	229.7	660	19000	2 46	11.75	2 700	0 0224	1000
50.8	274 3	1230	18000	2 33	11 03	3 030	0 0197	1000
63.3	325.1	2500	17000	2 21	9 12	2 960	0 0158	1000

Differentiating Equation [8], we obtain

$$d\epsilon = S\,dT + T\frac{\partial S}{\partial \sigma}\,d\sigma \quad\quad [13]$$

In the typical case of relaxation there is a direct relation between the increment $d\epsilon$ in the strain and the corresponding decrement in stress $d\sigma$. This may be written

$$d\epsilon = -\frac{1}{E}\,d\sigma \quad\quad [14]$$

if to the modulus of elasticity E is assigned a value appropriate to the structure involved. The maximum value of E is the modulus of elasticity of the material itself, and it is customary to use this value for most bolt problems. Introducing this value of $d\epsilon$ into Equation [13] and solving for $d\sigma$ the following result is obtained

$$d\sigma = -E\frac{S}{1 + TE\dfrac{\partial S}{\partial \sigma}}\,dT \quad\quad [15]$$

Using Equation [10] for S, this result becomes

$$d\sigma = -s_1\frac{e^{\sigma/s_1} - 1}{1 + Te^{\sigma/s_1}}\,dT \quad\quad [16]$$

An integration of Equation [16] with the boundary conditions $t = 0$, and $\sigma = \sigma_0$ would give the stress σ as a function of T. Through the empirical time function it is then possible to assign a value of t for every value of T.

There are no ready means for direct integration of Equation [15], or even Equation [16], so that step-by-step integration is necessary and far from formidable. The process is obvious and

[8] "Working Stresses in High-Temperature Service," by P. G. McVetty, *Mechanical Engineering*, vol. 56, 1934, p. 149.

[9] "The Creep Curve and Stability of Steels at Constant Stress and Temperature," by S. H. Weaver, Trans. A.S.M.E., vol. 58, 1936, paper RP-58-16, p. 745.

shown in Table 1 for the case of $\sigma_0 = 50{,}000$ lb per sq in., and $E = 30 \times 10^6$ lb per sq in. for the material covered by Fig. 1.

Fig. 4 shows the result of this calculation. It demonstrates the typical relaxation phenomena with an extremely rapid fall of the stress during the first hours.

The results shown in Fig. 4 are of interest in that they have been checked by direct relaxation tests conducted by Mochel.[10] The points shown represent test results obtained by Mochel for the same material at the same temperature. A great deal of emphasis cannot be placed on the relative agreement obtained, since the specimens in Mochel's tests were made from a different batch of material, and several years later, than those made by McVetty, but it lends some support to the soundness of the underlying theory. Additional relaxation tests are now planned.

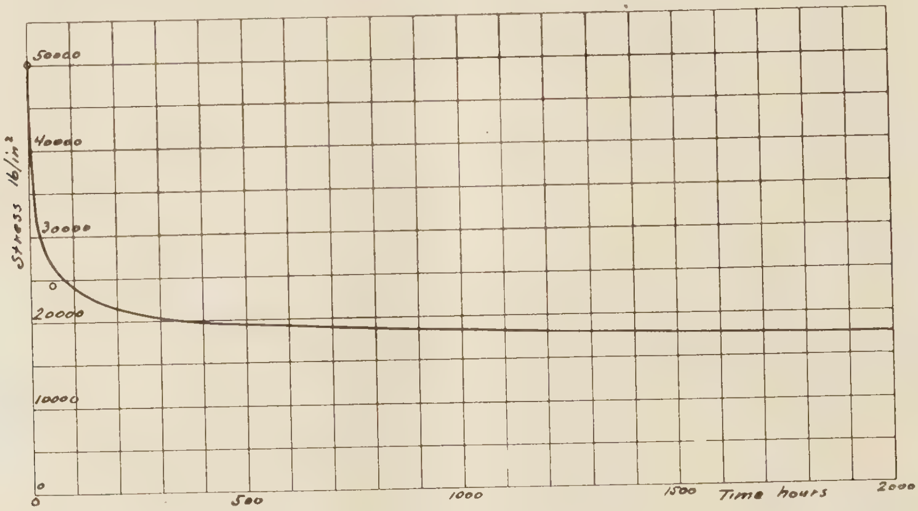

Fig. 4 Relaxation of 12 Per Cent Cr Steel at 850 F

For large values of the time, the function T may be replaced by $T = v_1 t$. If, in addition, e^{σ/s_1}, is large compared with unity, the relaxation proceeds approximately in accordance with the law

$$d\sigma = -s_1 \frac{dt}{t} \qquad [16a]$$

so that geometrical increments of time correspond to arithmetical increments in stress.

This phenomenon is approximated if a bolt, which has been in operation for an appreciable time, is retightened. The fundamental creep law, as given by Equation [9], indicates that it is immaterial whether or not the bolt was under stress during the first period of service. No reliable data exist so far which prove or disprove this deduction. Additional relaxation tests of the type described by Mochel[10] will be made to clear up this important question.

Two-Dimensional Problems. The fundamental equations of plastic flow may now be written for the general three-dimensional case. All that is needed is to regard the invariants e and s as related by the equation

$$e = ST \qquad [17]$$

where S is now a function of s alone, and T is a function of t alone. Introducing this relation into Equation [8], the equations of plastic flow become

$$\left.\begin{aligned} \epsilon_1 &= \frac{\partial s}{\partial \sigma_1} ST \\ \epsilon_2 &= \frac{\partial s}{\partial \sigma_2} ST \\ \epsilon_3 &= \frac{\partial s}{\partial \sigma_3} ST \end{aligned}\right\} \qquad [18]$$

These equations are not yet in a form which can be used for the solution of practical problems. They merely express the plastic strains as functions of the stresses and of time. To obtain solutions that are workable at all, it is usually necessary to so formulate the problems that a two-dimensional representation is possible. Moreover, it is a characteristic of those problems of machine design, for which creep is an important consideration, that they can usually be treated as cylinder problems. The two-dimensional representation of thin cylindrical shells is the simplest type of these problems; the radial stress distribution of thick rings is another. The following discussion will be restricted to these two cases. There are other problems of interest as well, but the general methods to be used will become apparent from the discussion of these two problems.

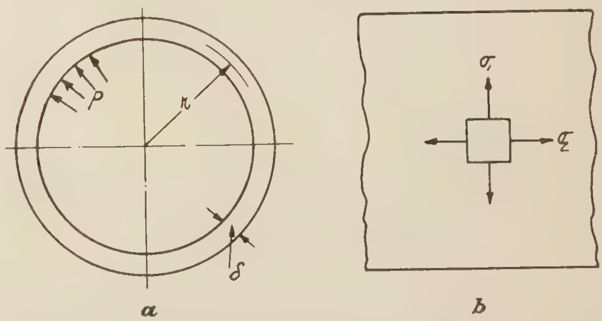

Fig. 5 Illustrating the Plastic Flow of Piping

Thin Cylindrical Shell. Consider a tube of mean radius r and thickness δ. The tangential stress is taken as σ_1, and the axial stress as σ_2, as indicated in Fig. 5. Piping under internal pressure

[10] A brief review of these tests will be given by N. L. Mochel at the Annual Meeting of The American Society of Mechanical Engineers, November 30–December 4, 1936, in connection with the report of the Joint A.S.T.M.-A.S.M.E. Research Committee on Effect of Temperature on the Properties of Metals.

is the most important practical form of this problem. If the internal pressure is denoted by p, the stresses are

$$\left.\begin{aligned} \sigma_1 &= \frac{pr}{\delta} = \sigma^* \\ \sigma_2 &= \frac{pr}{2\delta} = \frac{\sigma^*}{2} \\ \sigma_3 &= 0 \text{ (or small)} \end{aligned}\right\} \qquad [19]$$

The stress σ^* is the stress usually designated as the working stress of the pipe. We now have

$$\left.\begin{aligned} s &= \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2} = \frac{\sqrt{3}}{2}\sigma^* \\ \frac{\partial s}{\partial \sigma_1} &= \frac{2\sigma_1 - \sigma_2}{2s} = \frac{\sqrt{3}}{2} \\ \frac{\partial s}{\partial \sigma_2} &= \frac{2\sigma_2 - \sigma_1}{2s} = 0 \end{aligned}\right\} \qquad [20]$$

so that

$$\left.\begin{aligned} \epsilon_1 &= (\sqrt{3}/2)ST \\ \epsilon_2 &= 0 \end{aligned}\right\} \qquad [21]$$

Thus, the radial creep of the pipe is obtained directly from the creep curves of the material itself by using for the stress variable $(\sqrt{3}/2)\sigma^*$, and multiplying the strain by the factor $\sqrt{3}/2$. The axial creep is zero, so that the creep conditions are not influenced by any elastic restrictions in the axial direction, except in so far as they may cause stresses through thermal expansions.

Thick Cylinders and Disks. Space does not permit a discussion of the general methods by which two- or three-dimensional problems may be treated, nor have these methods been developed to a point where a treatment in terms of broad generalities is available. The characteristic feature of these problems is that for every increment of time there is not only a set of increments of plastic strains, but also a set of increments of the stresses themselves. A consideration of great practical importance is whether or not the flow will proceed in such a manner that the increments of the stresses approach zero. If this is the case, a stationary state of flow is eventually approached, which is usually characterized by very simple relations.

It is evident from these remarks that this class of problems suggests the use of the calculus of variations as far as the fundamental approach is concerned. The mathematical difficulties are often very great, and so far very little progress in the way of practical solutions has been made.

The following discussion, while applying specifically to thick cylinders and disks, outlines the essential principles involved in applying the variational method. For simplicity, the discussion has been restricted to the simplest case of a free disk, with a known condition of elastic stresses as the starting point. The axial stresses are assumed to be zero at all times, so that no elastic restraint in the axial direction is involved. The method requires only minor modification to be applicable to other cases, such as a disk shrunk on a shaft, or the bending of a beam.

The first step in the analysis is to establish a relation between increments in time or time function, strain, and stress. This relation is obtained by differentiation of the Equations [18] of plastic flow, of which only two need to be considered. The first equation, with index 1 for stress and strain, will apply to the tangential direction; the second equation, with index 2 will apply to the radial direction.

The invariant s takes the form

$$s = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2} \qquad [22]$$

As a matter of convenience the derivatives of this function will be given the following symbols

$$\left.\begin{aligned} A_1 &= \frac{\partial s}{\partial \sigma_1} = \frac{2\sigma_1 - \sigma_2}{2s} \\ A_2 &= \frac{\partial s}{\partial \sigma_2} = \frac{2\sigma_2 - \sigma_1}{2s} \end{aligned}\right\} \qquad [23]$$

$$\left.\begin{aligned} a_{11} &= \frac{\partial A_1}{\partial \sigma_1} = \frac{\partial^2 s}{\partial \sigma_1^2} = \frac{1 - A_1^2}{s} \\ a_{12} = a_{21} &= \frac{\partial A_1}{\partial \sigma_2} = \frac{\partial A_2}{\partial \sigma_1} = \frac{\partial^2 s}{\partial \sigma_1 \partial \sigma_2} = -\frac{\frac{1}{2} + A_1A_2}{s} \\ a_{22} &= \frac{\partial A_2}{\partial \sigma_2} = \frac{\partial^2 s}{\partial \sigma_2^2} = \frac{1 - A_2^2}{s} \end{aligned}\right\} \qquad [24]$$

The following combinations of these quantities will also be of use in the sequel

$$\left.\begin{aligned} b_{11} &= A_1^2 \frac{\partial S}{\partial s} + a_{11}S \\ b_{22} &= A_2^2 \frac{\partial S}{\partial s} + a_{22}S \\ b_{12} = b_{21} &= A_1A_2 \frac{\partial S}{\partial s} + a_{12}S \end{aligned}\right\} \qquad [25]$$

$$\left.\begin{aligned} M &= E(b_{11} + \gamma b_{12}) \\ N &= E(b_{12} + \nu b_{22}) \end{aligned}\right\} \qquad [26]$$

The fundamental equations of flow

$$\left.\begin{aligned} \epsilon_1 &= A_1ST \\ \epsilon_2 &= A_2ST \end{aligned}\right\} \qquad [27]$$

will now be differentiated, remembering that

$$dS = \frac{\partial S}{\partial s}[A_1 d\sigma_1 + A_2 d\sigma_2] \qquad [28]$$

and

$$\left.\begin{aligned} dA_1 &= a_{11}d\sigma_1 + a_{12}d\sigma_2 \\ dA_2 &= a_{12}d\sigma_1 + a_{22}d\sigma_2 \end{aligned}\right\} \qquad [29]$$

After rearrangement, the differentiation of Equations [27] gives the following result

$$\left.\begin{aligned} \Delta\epsilon_1 &= A_1S\,\Delta T + T(b_{11}\Delta\sigma_1 + b_{12}\Delta\sigma_2) \\ \Delta\epsilon_2 &= A_2S\,\Delta T + T(b_{12}\Delta\sigma_1 + b_{22}\Delta\sigma_2) \end{aligned}\right\} \qquad [30]$$

It will be found advantageous to combine these expressions into the following single equation

$$\frac{1 - \nu^2}{E}\Delta\sigma_1^* = (A_1 + \nu A_2)S\,\Delta T + \frac{T}{E}(M\,\Delta\sigma_1 + N\,\Delta\sigma_2) \qquad [31]$$

where $\Delta\sigma_1^*$ denotes the stress parameter

$$\Delta\sigma_1^* = \frac{E}{1 - \nu^2}(\Delta\epsilon_1 + \nu\,\Delta\epsilon_2) \qquad [32]$$

Consider Equation [31] as applying to a specific value of r. If a small ring element $\Delta\xi$ at $r = \xi$ be permitted to deform plastically, as indicated in Fig. 6a, certain internal stresses are set up throughout the entire ring. These internal stresses may be determined by direct application of the Lamé solution. If an element

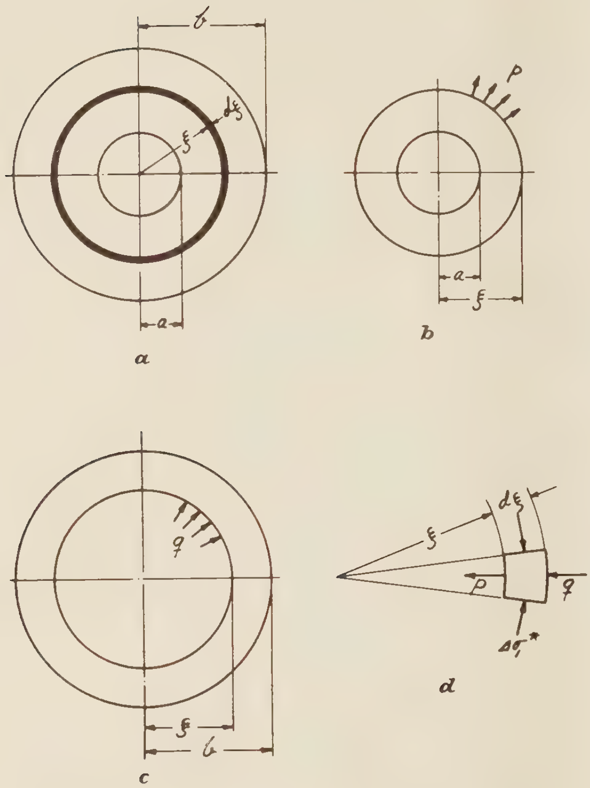

FIG. 6 ILLUSTRATING PLASTIC DEFORMATION OF A THICK RING

at $r = r$ deforms it will obtain a tangential stress of $-\Delta\sigma_1^*$ and a radial stress which is zero, or a differential of higher order. The total change of stress at $r = r$ is the sum of all the changes which are produced when ξ varies from a to r and from r to b. It is shown in the Appendix that at the point r

$$\left.\begin{aligned}\Delta\sigma_1 &= P\,\Delta\sigma_1^* \\ \Delta\sigma_2 &= Q\,\Delta\sigma_1^*\end{aligned}\right\} \dots\dots\dots\dots\dots\dots [33]$$

where P and Q are functions of r determined by

$$\left.\begin{aligned}P &= \frac{\Delta\xi}{2(b^2 - a^2)}\left[\xi\sum_r^b\left(1 + \frac{a^2}{r^2}\right)\frac{b^2+\xi^2}{\xi}k_r^\xi + \xi\sum_a^r\left(1+\frac{b^2}{r^2}\right)\frac{\xi^2+a^2}{\xi}k_r^\xi\right] - 1 \\ Q &= \frac{\Delta\xi}{2(b^2 - a^2)}\left[\xi\sum_r^b\left(1 - \frac{a^2}{r^2}\right)\frac{b^2+\xi^2}{\xi}k_r^\xi + \xi\sum_a^r\left(1-\frac{b^2}{r^2}\right)\frac{\xi^2+a^2}{\xi}k_r^\xi\right]\end{aligned}\right\} . [34]$$

The factor k_r^ξ is determined by

$$k_r^\xi = \frac{\Delta\sigma_{1\xi}^*}{\Delta\sigma_{1r}^*} \cong \frac{[(A_1 + \nu A_2)S]_{r\,=\,\xi}}{[(A_1 + \nu A_2)S]_{r\,=\,r}};\; k_r^r = k_\xi^\xi = 1 \dots [35]$$

and represents an approximation of the ultimate variation in ξ of $\Delta\sigma_1^*$. Combining Equations [31] and [33], the following expression is obtained for $\Delta\sigma_1^*$

$$\Delta\sigma_1^* = \frac{E(A_1 + \nu A_2)S}{1 - \nu^2 - T(MP + NQ)}\,\Delta T \dots\dots\dots [36]$$

which, in order to be acceptable, must check the assumed values of k_r^ξ.

This gives $\Delta\sigma_1^*$, and hence $\Delta\sigma_1$ and $\Delta\sigma_2$ as functions of r for the values of ΔT and T in question. These stress increments are now added to the original set of stresses (the elastic solution). The process is repeated for the next increment ΔT, and so on.

It is evident by the process selected that the stress increments obtained for each increment of T are in internal equilibrium, that is, they satisfy the boundary conditions of a free ring.

If, in addition, it is required to know the plastic deformations, they are most easily obtained directly from Equations [30] which become

$$\left.\begin{aligned}\Delta\epsilon_1 &= A_1 S\,\Delta T + T(b_{11}P + b_{12}Q)\,\Delta\sigma_1^* \\ \Delta\epsilon_2 &= A_2 S\,\Delta T + T(b_{12}P + b_{22}Q)\,\Delta\sigma_1^*\end{aligned}\right\} \dots\dots [37]$$

In most practical problems only the tangential deformations at the inner and outer radius are of interest. The radial displacement of any point at radius r is $r\,\Delta\epsilon_1$.

The results may be simplified somewhat further by taking for the stress function S the value proposed in Equation [10], namely

$$S = \frac{s_1}{E}\left(e^{s/s_1} - 1\right) \dots\dots\dots\dots\dots\dots [10]$$

Introducing the functions

$$F = e^{s/s_1};\; F' = \frac{e^{s/s_1} - 1}{s/s_1} \dots\dots\dots\dots [38]$$

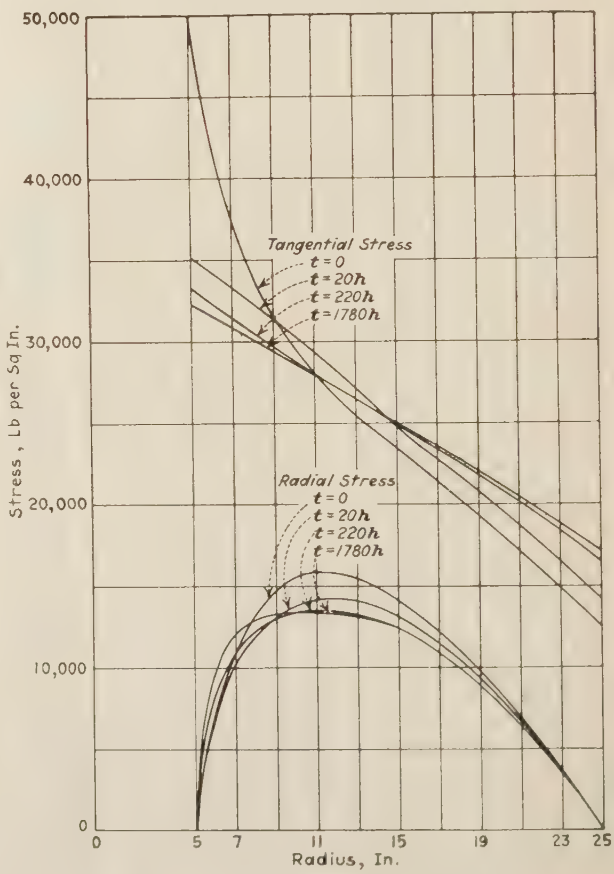

FIG. 7 STRESS DISTRIBUTION IN A DISK

TABLE II

Material: 12% CR. STEEL 850°F — PLASTIC FLOW OF DISK — Page 1

Item	r = 7	11	15	19	23
σ_1	37,250	28,090	23,360	19,230	14,850
σ_2	11,160	15,840	14,080	9,730	3,610
$(\sigma_2/\sigma_1)^2$	.090	.318	.364	.256	.059
σ_2/σ_1	.300	.564	.603	.506	.243
$1+(\sigma_2/\sigma_1)^2-\sigma_2/\sigma_1$	.790	.754	.761	.750	.816
S	33,100	24,400	20,360	16,650	13,410
S/S_1	4.29	3.16	2.64	2.16	1.74
F	73.0	23.6	14.01	8.67	5.70
F′	16.80	7.15	4.93	3.55	2.70
$\sigma_1-\sigma_2/2$	31,670	20,170	16,320	14,365	13,045
A_1	.956	.827	.801	.863	.973
$\sigma_2-\sigma_1/2$	-7465	1795	2400	115	3815
A_2	-.225	.074	.118	.007	-.284
νA_2	-.068	.022	.035	.003	-.085
$A_1+\nu A_2$	.888	.849	.836	.866	.888
$A_1(A_1+\nu A_2)$	.850	.702	.670	.748	.865
$A_2(A_1+\nu A_2)$	-.200	.063	.099	.008	-.252
$1-\frac{\nu}{2}-A_1(A_1+\nu A_2)$	0	.148	.180	.102	-.015
$\nu-\frac{1}{2}-A_2(A_1+\nu A_2)$	0	-.263	-.299	-.208	.052
$A_1(A_1+\nu A_2)F$	62.1	16.57	9.39	6.48	4.93
$A_2(A_1+\nu A_2)F$	-14.6	1.49	1.39	.07	-1.44
$[1-\frac{\nu}{2}-A_1(A_1+\nu A_2)]F'$	0	1058	888	362	-.040
$[\nu-\frac{1}{2}-A_2(A_1+\nu A_2)]F'$	0	-1882	-1475	-.739	.140
M	62.1	17.63	10.28	6.84	4.89
N	-14.6	-.39	-.08	-.67	-1.30
$1-\nu^2-T(MP+NQ)$	1.196	.846	.836	.823	.809
$S(A_1+\nu A_2)F'$	494,000	148,300	84,000	51,200	32,200
$\Delta\sigma^*$	12400	5260	3010	1865	1195
k_r^ξ, ξ = 7	1	3.33	5.88	9.65	15.35
ξ = 11	.300	1	1.77	2.90	4.60
ξ = 15	.170	.567	1	1.64	2.61
ξ = 19	.104	.345	.610	1	1.59
ξ = 23	.065	.217	.383	.628	1
$\Delta\sigma_1$	-4000	1270	1440	1537	1613
$\Delta\sigma_2$	-793	-1715	-968	-475	-130

$S_1 = 7710$	$E = 30\times10^6$	$\nu = 0.3$	$1-\nu^2 = .91$			
$a = 5$	$b = 25$	$\Delta\xi = 4$	$\frac{\Delta\xi}{2(b^2-a^2)} = \frac{1}{300}$			
$\Delta T = .03$	$T_{av} = .015$	$t = 0$	$t+\Delta t = 20$			
P and Q						
$\frac{\xi^2+b^2}{\xi}$	ξ	P, r = 7	11	15	19	23	$\frac{\xi^2+a^2}{\xi}$
	$1+b^2/r^2$	13.76	6.17	3.78	2.731	2.181	
96.3	7	145.4	65.2	39.95	28.85	23.05	10.56
		.484	.724	.784	.928	1.180	
67.8	11	102.4	81.9	50.2	36.25	28.95	13.27
		.102	.273	.296	.350	.444	
56.7	15	85.6	68.4	63.0	45.5	36.35	16.67
		.048	.129	.210	.249	.316	
51.9	19	78.4	62.7	57.7	55.5	44.3	20.32
		.027	.072	.117	.185	.235	
50.2	23	75.8	60.6	55.7	53.7	52.5	24.09
		.016	.044	.071	.112	.175	
	$1+a^2/r^2$	1.510	1.207	1.111	1.069	1.047	
	P	-.323	.242	.478	.824	1.350	
$\frac{\xi^2+b^2}{\xi}$		Q, r = 7	11	15	19	23	$\frac{\xi^2+a^2}{\xi}$
	$1-b^2/r^2$	-11.76	-4.17	-1.778	-.731	-.181	
96.3	7	-38.5	-44.1	-18.78	-7.72	-1.912	10.56
		-.128	-.484	-.368	-.248	-.098	
67.8	11	33.2	-.8	-23.6	-9.70	-2.40	13.27
		.033	-.003	-.139	-.094	-.037	
56.7	15	27.8	45.0	+10.4	-12.20	-3.02	16.67
		.016	.085	.035	-.067	-.026	
51.9	19	25.4	41.2	46.2	+16.7	-3.68	20.32
		.009	.047	.094	.056	-.020	
50.2	23	24.6	38.8	44.6	46.7	+21.7	24.09
		.005	.029	.057	.098	.072	
	$1-a^2/r^2$	.490	.793	.889	.931	.953	
	Q	-.064	-.326	-.321	-.255	-.109	

Item	r = 7	11	15	19	23
MP	-20.05	4.17	4.92	5.63	6.61
NQ	.93	.13	.03	.17	.14
MP+NQ	-19.12	4.30	4.95	5.80	6.75
T(MP+NQ)	-.286	.064	.074	.087	.101
A_1^2F	66.8	16.15	9.00	6.47	5.39
$(1-A_1^2)F'$	1.45	2.25	1.77	.91	.14
$b_{11}E$	68.25	18.4	10.77	7.38	5.53
A_1A_2F	-15.7	1.45	1.32	.07	-1.58
$(\frac{1}{2}+A_1A_2)F'$	4.8	4.01	2.93	1.81	.60
$b_{12}E$	-20.5	-2.56	-1.61	-1.74	-2.18
$Pb_{11}E$	-22.0	4.45	5.15	6.08	7.47
$Qb_{12}E$	1.3	.83	.52	.44	.24
$E(Pb_{11}+Qb_{12})$	-20.7	5.28	5.67	6.52	7.71
$TE(Pb_{11}+Qb_{12})\Delta\sigma_1^*$	-3850	417	256	182	138
SA_1F'	532,000	144,200	80,500	51,100	35,300
$E\,\Delta\rho/r : \Delta T$	528,000	144,600	80,800	51,300	35,400
$\Delta\rho\cdot10^3$	3.70	1.59	.21	0.98	0.81

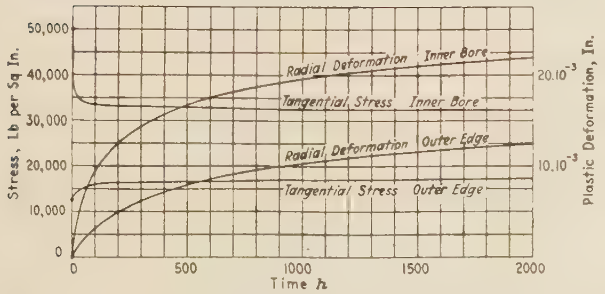

FIG. 8 CHANGE OF STRESS AND PLASTIC DEFORMATION IN A DISK

we then obtain

$$S = \frac{s}{E}F'; \quad \frac{\partial S}{\partial s} = \frac{1}{E}F \quad \ldots\ldots [39]$$

$$\left.\begin{aligned} b_{11} &= \frac{1}{E}[A_1^2F + (1 - A_1^2)F'] \\ b_{22} &= \frac{1}{E}[A_2^2F + (1 - A_2^2)F'] \\ b_{12} = b_{21} &= \frac{1}{E}[A_1A_2F - (1/2 + A_1A_2)F'] \end{aligned}\right\} \ldots [40]$$

$$\left.\begin{aligned} M &= A_1(A_1+\nu A_2)F + [1 - \nu/2 - A_1(A_1+\nu A_2)]F' \\ N &= A_2(A_1+\nu A_2)F + [\nu - 1/2 - A_2(A_1+\nu A_2)]F' \end{aligned}\right\} \ldots [41]$$

and

$$\Delta\sigma_1^* = \frac{s(A_1+\nu A_2)F'}{1-\nu^2-T(MP+NQ)}\Delta T \quad \ldots\ldots [42]$$

Fig. 7 shows the results of a calculation which has been carried out in this manner for a free rotating disk, the inside and outside radii being 5 and 25 in., respectively. The speed of rotation was selected in such a manner as to give a tangential stress of 50,000 lb per sq in. at the inner bore in the elastic state. The material is that covered by Fig. 1. These results indicate that for practical purposes a stationary state of stress may be considered to exist after the lapse of about 1000 hr. This state of stress satisfies the conditions established for the steady state in the Appendix.

Fig. 8 shows the changes with time in radial stress and plastic deformation at the inner and outer bore. The initial fall of the radial stress at the inner bore is similar to that shown in Fig. 3 for pure relaxation, but the reduction for subsequent periods of time is much smaller and eventually it becomes zero.

Table 2 illustrates the extent of the calculations required for one of the increments of the time function T.

Appendix

Assume that an element $d\xi$ of the ring shown in Fig. 6a at radius ξ is given plastic deformations $\Delta\epsilon_1$ and $\Delta\epsilon_2$ in the tangential and the radial directions, respectively. If the element is small the radial deformation of the ring is a differential of a higher order, and the element itself is subjected to a tangential stress

$$\Delta\sigma_1 = -\frac{E}{1-\nu^2}(\Delta\epsilon_1 + \nu\,\Delta\epsilon_2) = -\Delta\sigma_{1\xi}^* \quad \ldots\ldots [43]$$

The radial stress, indicated by Fig. 6d, will be denoted by p at the inner edge, and $-q$ at the outer edge of the element. The equilibrium of the element itself requires that

$$p + q = \Delta\sigma_{1\xi}^*\frac{d\xi}{\xi} \quad \ldots\ldots [44]$$

If the ring element is now removed, equilibrium may be maintained by the addition of the radial stress p on the inner portion and $-q$ on the outer portion, as indicated in Figs. 6b and 6c, respectively. The stresses in the inner and outer portions of the ring are obtained directly from the Lamé solution

$$\left.\begin{aligned} d(\Delta\sigma_1)_r &= \frac{\xi^2}{\xi^2 - a^2}\left(1 + \frac{a^2}{r^2}\right)p \\ d(\Delta\sigma_2)_r &= \frac{\xi^2}{\xi^2 - a^2}\left(1 - \frac{a^2}{r^2}\right)p \end{aligned}\right\}_{r<\xi} \quad \ldots\ldots[45]$$

$$\left.\begin{aligned} d(\Delta\sigma_1)_r &= \frac{\xi^2}{b^2 - \xi^2}\left(1 + \frac{b^2}{r^2}\right)q \\ d(\Delta\sigma_2)_r &= \frac{\xi^2}{b^2 - \xi^2}\left(1 - \frac{b^2}{r^2}\right)q \end{aligned}\right\}_{r>\xi} \quad \ldots\ldots\ldots[46]$$

Considerations of continuity demand that the tangential stress at $r = \xi$ be the same for both expressions. This establishes the ratio of $p{:}q$ which, when introduced into Equation [44] gives

$$\left.\begin{aligned} p &= \frac{(\xi^2 - a^2)(b^2 + \xi^2)}{2\xi^2(b^2 - a^2)}\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \\ q &= \frac{(b^2 - \xi^2)(\xi^2 + a^2)}{2\xi^2(b^2 - a^2)}\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \end{aligned}\right\} \quad \ldots\ldots\ldots[47]$$

Equations [45] and [46] now become

$$\left.\begin{aligned} d(\Delta\sigma_1)_r &= \frac{1 + \frac{a^2}{r^2}}{2(b^2 - a^2)}(b^2 + \xi^2)\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \\ d(\Delta\sigma_2)_r &= \frac{1 - \frac{a^2}{r^2}}{2(b^2 - a^2)}(b^2 + \xi^2)\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \end{aligned}\right\}_{r<\xi} \quad \ldots\ldots[48]$$

$$\left.\begin{aligned} d(\Delta\sigma_1)_r &= \frac{1 + \frac{b^2}{r^2}}{2(b^2 - a^2)}(\xi^2 + a^2)\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \\ d(\Delta\sigma_2)_r &= \frac{1 - \frac{b^2}{r^2}}{2(b^2 - a^2)}(\xi^2 + a^2)\,\Delta\sigma_{1\xi}^*\,\frac{d\xi}{\xi} \end{aligned}\right\}_{r>\xi} \quad \ldots\ldots[49]$$

and establish the change in stress at $r = r$ incidental to the deformations $\Delta\epsilon_1$ and $\Delta\epsilon_2$ (expressed through $\Delta\sigma_{1\xi}^*$), at $r = \xi$.

The total change in stress at the point $r = r$ is now obtained by integration of these expressions, remembering that the element at $r = \xi$ gives an increment to the tangential stress of $-\Delta\sigma_{1r}^*$. This gives

$$\left.\begin{aligned} \Delta\sigma_{1r} &= -\Delta\sigma_{1r}^* + \int_r^b \frac{1 + \frac{a^2}{r^2}}{2(b^2 - a^2)}\,\frac{b^2 + \xi^2}{\xi}\,\Delta\sigma_{1\xi}^* d\xi \\ &\quad + \int_a^r \frac{1 + \frac{b^2}{r^2}}{2(b^2 - a^2)}\,\frac{\xi^2 + a^2}{\xi}\,\Delta\sigma_{1\xi}^* d\xi \\ \Delta\sigma_{2r} &= \int_r^b \frac{1 - \frac{a^2}{r^2}}{2(b^2 - a^2)}\,\frac{b^2 + \xi^2}{\xi}\,\Delta\sigma_{1\xi}^* d\xi \\ &\quad + \int_a^r \frac{1 - \frac{b^2}{r^2}}{2(b^2 - a^2)}\,\frac{\xi^2 + a^2}{\xi}\,\Delta\sigma_{1\xi}^* d\xi \end{aligned}\right\} \quad \ldots[50]$$

The difficulty in the way of integration is due to the fact that $\Delta\sigma_{1\xi}^*$ under the integration signs is a complicated function of ξ to be determined from the expression

$$\Delta\sigma_{1\xi}^* = \frac{E}{1 - \nu^2}\left[(A_1 + \nu A_2)S\,\Delta T + \frac{T}{E}(M\,\Delta\sigma_1 + N\,\Delta\sigma_2)\right] \quad \ldots\ldots\ldots[51]$$

where all quantities are now to be regarded as functions of ξ. To overcome this difficulty the approximation is made that under the integrals in Equation [50] $\Delta\sigma_{1\xi}^*$ varies with ξ as it would in Equation [51] if $\Delta\sigma_1$ and $\Delta\sigma_2$ were zero. The simplest way of injecting this variation is to introduce the factor

$$k_r^\xi \cong \frac{[(A_1 + \nu A_2)S]_{r\,=\,\xi}}{[(A_1 + \nu A_2)S]_{r\,=\,r}} \quad \ldots\ldots\ldots\ldots\ldots[52]$$

so that

$$k_r^r = k_\xi^\xi = 1 \quad \ldots\ldots\ldots\ldots\ldots\ldots[53]$$

Introducing the functions

$$\left.\begin{aligned} P &= \int_r^b \frac{1 + \frac{a^2}{r^2}}{2(b^2 - a^2)}\,\frac{b^2 + \xi^2}{\xi}\,k_r^\xi d\xi \\ &\quad + \int_a^r \frac{1 + \frac{b^2}{r^2}}{2(b^2 - a^2)}\,\frac{\xi^2 + a^2}{\xi}\,k_r^\xi d\xi - 1 \\ Q &= \int_r^b \frac{1 - \frac{a^2}{r^2}}{2(b^2 - a^2)}\,\frac{b^2 + \xi^2}{\xi}\,k_r^\xi d\xi \\ &\quad + \int_a^r \frac{1 - \frac{b^2}{r^2}}{2(b^2 - a^2)}\,\frac{\xi^2 + a^2}{\xi}\,k_r^\xi d\xi \end{aligned}\right\} \quad \ldots[54]$$

we obtain

$$\left.\begin{aligned} \Delta\sigma_{1r} &= P\,\Delta\sigma_{1r}^* \\ \Delta\sigma_{2r} &= Q\,\Delta\sigma_{1r}^* \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots[55]$$

and from Equation [51]

$$\Delta\sigma_{1r}^* = \frac{E(A_1 + \nu A_2)S}{1 - \nu^2 - T(MP + NQ)}\,\Delta T \quad \ldots\ldots[56]$$

which is the solution of the problem.

For practical calculations, the quantity k_r^ξ must be approximated by selecting the functional relation obtained from a previous step of the calculation. The value given by Equation [52] is appropriate for the first step, where the elastic solution gives the functional relationship. For subsequent steps, k_r^ξ is best evaluated from the previous step in the calculation. Thus, if $\Delta\sigma_{1r}^* = G(r)$ is the result of the previous step, then

$$k_r^\xi = \frac{G(\xi)}{G(r)} \quad \ldots\ldots\ldots\ldots\ldots\ldots[52a]$$

may be used for the subsequent step, and so on.

In general it is to be expected that every increment of time will produce a set of increments in stress as well as a set of increments in strain. Certain conditions may be postulated, however, under which the flow may continue, but the stresses remain stationary. Such a condition of stress would be classified as a stationary state.

It is evident from the preceding discussion that this stationary state is characterized by $\Delta\sigma_1 = 0$ and $\Delta\sigma_2 = 0$. This may be ob-

tained if $\Delta\sigma_1^* = 0$ or if $P = 0$ and $Q = 0$. The former case would interfere with the fixed boundary conditions and must be ruled out. The latter alternative is obtained from Equations [54] in the form of two simultaneous integral equations for k_r^ξ. If Equation [52a] is used for this quantity, the integral equations take the form

$$\left.\begin{aligned} &\left(1+\frac{a^2}{r^2}\right)\int_r^b \frac{b^2+\xi^2}{\xi}G(\xi)d\xi + \left(1+\frac{b^2}{r^2}\right)\int_a^r \frac{\xi^2+a^2}{\xi} G(\xi)d\xi = 2(b^2-a^2)G(r) \\ &\left(1-\frac{a^2}{r^2}\right)\int_r^b \frac{b^2+\xi^2}{\xi}G(\xi)d\xi + \left(1-\frac{b^2}{r^2}\right)\int_a^r \frac{\xi^2+a^2}{\xi} G(\xi)d\xi = 0 \end{aligned}\right\} \quad \text{[57]}$$

These equations may be solved for the two integrals. Since these integrals do not contain r, ξ may be replaced by r. The solution gives

$$\left.\begin{aligned} \int_r^b \frac{b^2+r^2}{r} G(r)dr &= (b^2-r^2)G(r) \\ \int_a^r \frac{r^2+a^2}{r} G(r)dr &= (r^2-a^2)G(r) \end{aligned}\right\} \quad \text{[58]}$$

The question of the possibility of a stationary state of stress is now reduced to a search for a function $G(r)$ which will satisfy these two integral equations. An inspection shows that they are satisfied by

$$G(r) = \Delta\sigma_1^* = \frac{c}{r} \quad \text{[59]}$$

where c is a constant in the sense that it does not vary with r. When this state of stress has been reached, $\Delta\sigma_1$ and $\Delta\sigma_2$ will be zero, and they will remain zero as the flow continues. Equation [36] now shows that the flow will proceed in accordance with the law

$$\frac{d\sigma_1^*}{dT} = \frac{E}{1-\nu^2}(A_1+\nu A_2)\,S = \frac{c}{r} \quad \text{[60]}$$

Equation [32] gives the corresponding expressions for $d\epsilon_1/dT$ and $d\epsilon_2/dT$. If these quantities are written ζ'/r and $d\,\zeta'/dr$, the following differential equation is obtained from Equations [32] and [60]

$$\frac{\zeta'}{r} + \nu\frac{d\zeta'}{dr} = \frac{c_1}{r} \quad \text{[61]}$$

where $c_1 = c\,(1-\nu^2)/E$.

After integration this gives

$$\left.\begin{aligned} \frac{d\epsilon_1}{dT} &= \frac{\zeta'}{r} = A_1S = \frac{c_1}{r} + \frac{c_2}{r^{\frac{1+\nu}{\nu}}} \\ \frac{d\epsilon_2}{dT} &= \frac{d\zeta'}{dr} = A_2S = -\frac{1}{\nu}\frac{c_2}{r^{\frac{1+\nu}{\nu}}} \end{aligned}\right\} \quad \text{[62]}$$

where c_2 is another constant.

If, as is usually the case when the steady state has been reached, the time function T may be replaced by a straight line $T = v_1t$, these equations become

$$\left.\begin{aligned} \frac{d\epsilon_1}{dt} &= v_1\frac{\zeta'}{r} = \frac{\dot\zeta}{r} = v_1A_1S = v_1\left[\frac{c_1}{r} + \frac{c_2}{r^{\frac{1+\nu}{\nu}}}\right] \\ \frac{d\epsilon_2}{dt} &= v_1\frac{d\zeta'}{dr} = \frac{d\dot\zeta}{dr} = v_1A_2S = -\frac{v_1}{\nu}\frac{c_2}{r^{\frac{1+\nu}{\nu}}} \end{aligned}\right\} \quad \text{[63]}$$

where $\dot\zeta$ now represents ordinary differentiation with regard to the time.

To this condition of flow there now corresponds a state of stress which, once it is reached, will not alter. This state of stress must satisfy the elementary conditions of equilibrium as well as the boundary conditions, but it is not dependent upon the conditions of internal stress which the ring happened to possess at the beginning of the flow.

So far no method has been devised whereby this distribution of stress may be written in explicit form. The usual method so far has been to make a few step-by-step calculations, which soon lead to a close approximation of the steady state. The proof of the existence of a steady state is of great importance, however, not only because it will make the tedious step-by-step calculations unnecessary, but also because it permits a more exact determination of the stresses which ultimately will be reached.

The Creep Curve and Stability of Steels at Constant Stress and Temperature

By S. H. WEAVER,[1] SCHENECTADY, N. Y.

The creep rate for steels at constant stress and elevated temperature is composed of (1) the strain hardening with a rate which varies inversely with time, and (2) an asymptotic constant creep rate. The sum gives the creep rate at any time, and the formula for the total creep-extension curve.

For steels with a physical or structural change during creep, a third quantity must be added to the creep-extension formula, the test value of the quantity indicating the degree of instability of the test metal.

To illustrate four of the principal types of instability, the author presents four sets of long-time creep tests extending from 5500 hr to 5 yr. They represent changes due to (1) carbide spheroidization, (2) ferritic banding, (3) dendrites, and (4) alloy segregations.

IN PRESENTING the subject there is proposed a creep-time formula for metals, and a check on the equations is made with many long-time creep tests which extend over various periods of time from 1000 hr to 5 yr. Some of the tests, particularly those for the longer time, are still running.

The physicist has been interested in creep theory and tests ever since Sir William Thomson (Lord Kelvin) suggested the viscosity of solids. About 1910 Andrade (1)[2] in a study of plastic and viscous flow of metal experimented with wires of copper, iron, lead, tin, and frozen mercury. The extension-time curves for constant loads and temperatures had the same shape as the familiar creep-time curves for highly stressed steel.

Andrade described the initial permanent extension where the velocity of flow decreased with time as plastic flow. When the variable rate decreased later to a constant value, the phenomenon was termed viscous flow. In this paper the plastic action will be called "strain hardening" and the viscous-flow rate termed a constant creep rate for constant stress and temperature.

The basis of the proposed creep formulas is that the metals and alloys when held above certain constant temperatures and stresses, permanently elongate. The extension begins with a relatively high velocity or creep rate which quickly decreases in inverse proportion to time and approaches a constant or asymptotic-creep rate. These conditions, as shown in Appendix 1 lead to the creep-extension formula for constant stress and temperature

$$\epsilon = v_a t + a \log_{10} t - b \qquad [1]$$

$$b = a \log_{10} t_0 \qquad [2]$$

where ϵ = total creep elongation, in. per in. of length; v_a = constant creep rate, 10^{-7} in. per in. per hr (10^{-7} equals 1 per cent creep in 100,000 hr); a = strain-hardening constant; t = time, hr; t_0 = test constant indicating stability of metal structure and is a "time modulus" for the rate of strain hardening; and b = test constant for a particular test and material, and is considered as zero for stable metals.

The Constants in Creep

The creep characteristics of a given material at constant stress and temperature are defined by the asymptotic constant-creep rate v_a and the strain hardening constant a. There is an equally important stability constant t_0 which indicates the condition or change in condition of the test sample during creep test. This conclusion is based on the following reasoning: After passing Equation [1] through three test points, say the 200-, 500-, and 1000-hr points on the creep-time test curve, project the calculated creep curve toward zero on the time scale and the time axis is cut at some value t_0 given by Equation [2]. For a metal with unchanging structure during creep t_0 should be 1 to 3 hr, and b near zero. These quantities are sensitive due to the high velocity of the initial creep, and allowance must be made for small errors in the readings for zero-creep elongation and in time for the temperature and load adjustment. Experience has shown that t_0 can range from 1 to 3 hr and the material not show changing characteristics even in the longer-time tests. But when t_0 is greater than 3 hr for the 1000-hr test, one can find a slowly changing material or an unstable condition. Chrome-ferrite streamers in an alloy steel gradually broke up in a long-time test but their presence was indicated by t_0 of 4 to 8 hr. Heavy ferrite bands in an annealed rolled steel began creep with erratic or wavy curves; but when the banding was partly masked by heat-treatments, t_0 ranged from 10 to 20 hr. The cast alloy steels gave t_0 values from 2 to 50 hr, depending upon the before-test condition of the material with respects to dendrites and extent of anneals. In the tests referred to later, the character and evidence of unstable changes in the steels will be noted.

Application

When creep tests at constant stress and temperature are plotted using linear-scale coordinates, the first 1000 hr contains a knee in the curve which sometimes nearly reaches to a 90-deg bend as shown in Fig. 1. When the creep curve is plotted on semi-log paper as shown in Fig. 2, the knee has disappeared and that part of the curve approaches a straight line. An inspection of Equation [1] shows all quantities in the equation represent straight lines except $v_a t$ which causes the slight curvature in the early part of the curves when plotted on a semi-logarithmic scale. In practice, the first 1000 hr of a creep test at constant stress are plotted on a larger creep-extension scale than that shown in Fig. 2, after which a draftsman's logarithmic

[1] Turbine Engineering Department, General Electric Co., Schenectady, N. Y. Mr. Weaver was graduated from Purdue University in electrical engineering in 1903. Since that time he has been employed continuously by the General Electric Company, Schenectady Works in various departments of design and engineering. For the past six years he has been in charge of the metallurgical section of the turbine-engineering department.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Main Research Committee for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., November 30 to December 4, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

curve is used to fit a line to the test points. This fitting presents no difficulty as this curve is usually the smoothest one that can be drawn through the test points. Readings of total creep and time are then made from the curve at 200, 500, and 1000 hr (other time values can be used), and, when substituted in Equation [1], give three simultaneous equations with which the values of v_a, a, and b are found. Equation [2] can then be used to calculate t_0. This method determines the value of v_a and a from the difference in total creep between the first and second points and

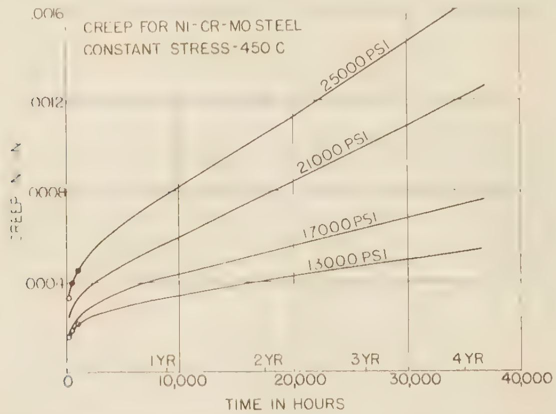

Fig. 1 Creep for Ni-Cr-Mo Steel at Constant Stress and 842 F (450 C)

between the second and third points, and, therefore, v_a and a are independent of the zero in the time-elongation test readings.

Tests

As the proposed formula for the creep curve is based on the hypothesis that the velocity of the strain-hardening extension varies inversely with time, the validity of the formula must be supported by test results.

The data published by Andrade (1) cover tests on wires of lead, tin, copper, and iron at different constant stresses and temperatures and are presented in the form of curves, tables and "viscosity constants," or constant-creep rates. All of his curves and tables can be accurately duplicated for the entire lengths by Equation [1], including the copper curve in his Table 3 where no "final viscous flow" was found, because the test ended before completion of the strain hardening.

The General Electric Company is conducting many long-time creep tests at constant loads and temperatures on alloyed steels. The creep formula was used on 72 constant-stress tests of 1000 to 2000 hr at temperatures of 752, 842, 932, 1022, and 1112 F. The equations applied to the curves at all of the temperatures giving the constant-creep rate, the strain hardening constant, and the stability constant.

It was usually difficult to obtain a smooth semilog curve through the test points below 200 hr. Comparing with some shorter-time higher-rate tests, which could be plotted down to 1 hr, it would seem that for apparently uniform metal structures there is required a small amount of creep extension to adjust the stress distribution within the test-metal section.

The strain-hardening constant a, for all of the temperatures previously mentioned, ranged from 0.1 to 0.3 $\times$ 10^{-3} when the creep rate was near 1 per cent, that is, for v_a = 0.5 to 5.0 $\times$ 10^{-7} in. per in. per hr. The unstable structures, the fine-grained oil-quenched steels, and all of the 1112-F tests tended toward the higher values of a for a given creep rate. For 1022 F and higher, the air-cooled and tempered steels had a structural formation, and strain-hardened rapidly, some in 150 to 450 hr, then extended at an artificial constant-creep rate for the remainder of the 1000 hr. The same steels when fully annealed were stable with normally shaped creep curves for the 1000 hr and gave a final higher creep strength than the air-cooled treatment after 3000-hr tests.

In addition to these tests, the formula was applied to 39 constant-stress creep curves of 5000 hr to 5 yr duration at 842 to 1022 F. In these tests, the data contained in the first 1000 hr were used to project the creep curve to the end of the test for a check on the accuracy of formula. When the estimated creep curve and final creep did not agree with test results, a larger stability constant was always found. A search would produce evidence of a structural or metallurgical change occurring in the steel.

The test constant t_0 served as a sensitive indicator of stability of metal structure, but it cannot foretell the slow change of carbide spheroidization due to the temperature of test. Dendritic cast steel not thoroughly heat-treated, rolled steel with ferritic banding, duplex structure of uneven grain size, and nonhardening steels containing alloy segregations are all subject during creep to changes in the structure of the steel which are detected by the t_0 value.

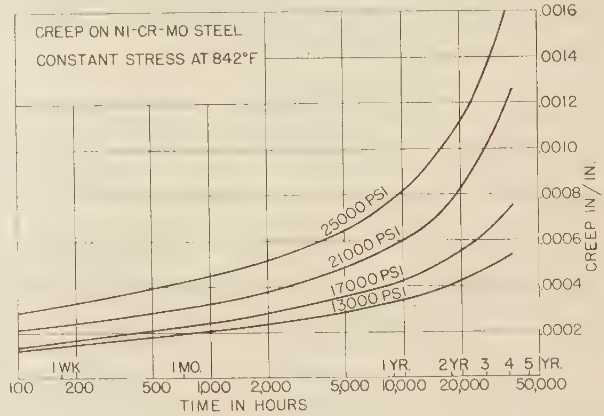

Fig. 2 Creep for Ni-Cr-Mo Steel at Constant Stress and 842 F (450 C) Plotted Against Logarithm of Time

A number of long-time tests will be presented in detail later. The tests were selected to illustrate the creep formula with special attention to the various types of structural changes in the steels. It must be understood that the long-time creep tests are presented here for their scientific interest only and are not representative of the practice of the General Electric Company in the use of materials for elevated temperature. While some of these steels have excellent properties, none of the tests presented fulfill all the present requirements or specifications for the chemical composition, fabrication, and heat-treatment used at the present time.

Long-Time Tests

The apparatus and methods of tests have been described by F. P. Coffin and T. H. Swisher (2). The operation and design of later furnaces have been described by P. H. Clark and E. L. Robinson (3).

The creep data are presented in the form of tables where three points within the first 1000 hr and one point at the end of the test are recorded as "test" values of total creep per unit length. The first 1000-hr data are used to calculate the constants by Equation [1] which are tabulated as "calculated." The total creep at the end of test is calculated with these constants in Equation [1] and checked against the tested value. In all of the cases of agreement with the final test point, the calculated creep

TABLE 1 CREEP-TEST DATA FOR NICKEL-CHROMIUM-MOLYBDENUM STEEL[a] AT 842 F (450 C)

Item no.	281	282	283	284
Test data:				
Stress, lb per sq in.	25000	21000	17000	13000
ϵ at 200 hr. ($\times 10^{-3}$)	0.330	0.243	0.173	0.157
ϵ at 500 hr. ($\times 10^{-3}$)	0.395	0.291	0.214	0.188
ϵ at 1000 hr. ($\times 10^{-3}$)	0.451	0.332	0.247	0.213
ϵ_1 at t_1 hr. ($\times 10^{-3}$)	1.125	0.813	0.555	0.423
t_1, hr.	20000[b]	20000[b]	20000[b]	20000[b]
Calculated data:				
v_a ($\times 10^{-7}$)	0.2530	0.1800	0.1023	0.0602
a ($\times 10^{-3}$)	0.1445	0.1064	0.0930	0.0736
b ($\times 10^{-3}$)	0.0078	0.0051	0.0130	0.0141
t_0	1.1300	1.1200	1.3800	1.5500
ϵ_1 at t_1 hr. ($\times 10^{-3}$)	1.1200	0.8120	0.5720	0.4230

[a] Analysis: 2.05 Ni, 0.83 Cr, 0.54 Mn, 0.45 Mo, and 0.31 C.
[b] Test still running.
NOTE: Heat-treatment consisted of furnace cooling from 825 C, oil quenching from 675 C, and slow cooling from 425 C.
NOTE: From a log-log graph at 842 F (450 C) with v_a = 1 per cent in 100,000 hr, σ = 47,000 lb per sq in., $a = 0.26 \times 10^{-3}$, $v_a = 0.035 \times 10^{-7} (\sigma/10{,}000)^{2.14}$, and $a = 0.055 \times 10^{-3}(\sigma/10{,}000)^{1.0}$.

curve represents the smoothest curve that can be drawn through the test points and is evidence of the correctness of Equation [1].

Results of creep test of five-years' duration, and which is still running is presented in Figs. 1 and 2. The material used in the tests consists of four bars of forged nickel-chromium-molybdenum steel which were heat-treated by furnace cooling through

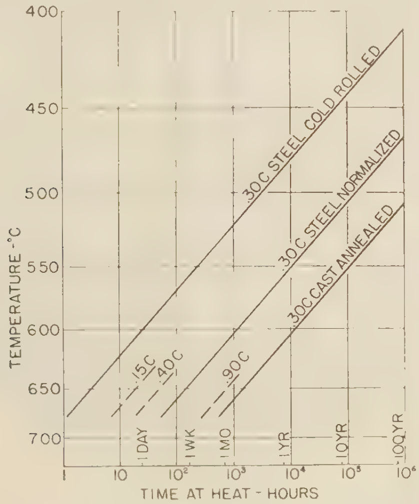

FIG. 3 TIME REQUIRED TO SPHEROIDIZE CARBON STEEL
(Normalized curve is for cast, forged, or cold-rolled steel heated to 900 C and air cooled. The differences in the dashed and solid curves are due to the degree of spheroidization used as a standard. Spheroidization increases the creep rate ten times. Curves drawn from data published by Bailey and Roberts (4).)

the critical temperature of the steel. The microstructure before test was not obtained on these four bars but a supposedly duplicate bar shows a structure that approaches a normalized or air-cooled steel. Table 1 records the chemical analysis, heat-treatment and the creep-test results on the four bars or test items. From 100 to 20,000 hr, the test and formula give an almost perfect agreement. Beyond 20,000 hr, there begins a slow separation of the calculated and tested creep curves, the tested values becoming the larger. The four items pass a minimum creep rate in 20,000 to 23,000 hr. At 37,000 hr the tested total creeps are 6 to 9 per cent higher, and the actual creep rates are 22 to 26 per cent greater than the calculated values. Seeking an explanation for this action involved consideration of many causes and close observation of the test for 1 to 2 yr. Eliminating all the suggested causes, including actual surface oxidation and thermocouple deterioration, there remained only the effect of slow carbide spheroidization in the steel.

TABLE 2 CREEP-TEST DATA FOR ANNEALED HOT-ROLLED STEEL BAR[a] AT 857 F (458.3 C)

Item no.	607	608	611	610	612
Test data:					
Stress, lb per sq in.	29068	21045	17032	13018	9005
ϵ at 200 hr. ($\times 10^{-3}$)	...	...	...	...	...
ϵ at 500 hr. ($\times 10^{-3}$)	...	...	...	...	...
ϵ at 1000 hr. ($\times 10^{-3}$)	15.6	3.42	1.62	0.905	0.470
ϵ_1 at t_1 hr. ($\times 10^{-3}$)	39.0	11.75	8.60	4.040	1.440
t_1 = end of test, hr.	5000	7500	14000	14000	14000
Calculated data:					
v_a ($\times 10^{-7}$)	55.8	12.40	4.83	2.05	0.555
a ($\times 10^{-3}$)	0.670	0.480	0.388	0.300	0.206
b ($\times 10^{-3}$)	−8.600	−0.740	0.025	0.200	0.204
t_0	...	...	...	...	...
ϵ_1 at t_1 hr. ($\times 10^{-3}$)	39.000	11.90	8.340	3.920	1 430

[a] Analysis: 1.78 Ni, 0.77 Cr, 0.55 Mn, 0.36 Mo, and 0.37 C.
NOTE: Heat-treatment consisted of furnace cooling from 825 C.
NOTE: From a log-log graph at 857 F (458.3 C) with v_a = 1 per cent in 100,000 hr, the stress σ = 10,800 lb per sq in., $a = 0.25 \times 10^{-3}$, $v_a = 0.74 \times 10^{-7}(\sigma/10{,}000)^{3.83}$, and $a = 0.23 \times 10^{-3}(\sigma/10{,}000)^{1.0}$.

Spheroidization in carbon steel has been investigated by Bailey and Roberts (4). In the pearlitic grains of an annealed steel, when viewed at a high magnification, the iron carbide appears in a lamellar formation. Upon prolonged heating, the carbide laminations coalesce into minute globular forms. Bailey and Roberts found that the time required for this change and the temperature were related by the law common to the time-temperature for many physical changes and chemical reactions, that the time for change is not influenced by the presence of a creep stress, and that the effect of a fully changed structure was to increase the creep rate approximately tenfold. Their formula and tests, abridged in Fig. 3, show that the time required to spheroidize a carbon steel at 450 C is between 80 and 300 yr. The variation in time is due to the long-range projection of the general time law from tests over the 1- to 1000-hr period. In the 2 yr after the minimum creep-rate points in the tests listed in Table 1, the effect of spheroidization upon the creep rate would be approximately [(10 — 1)2]/80 = 22.5 per cent increase, which compares with 22 to 26 per cent for the tests. This carbide-spheroidization factor must be so slow-working at 842 F (450 C) that the influence cannot be measured until after 2 yr. Therefore, it does not appear to invalidate the proposed creep formula, but rather is an element to add to the formula for very long tests or for higher-temperature service conditions. It should be particularly noted that the steel in Table 1 was tested for 5 yr at constant stresses and at 842 F (450 C), and that the creep test indicates the unusual creep strength of 47,000 lb per sq in. for an asymptotic or constant-creep rate of 1 per cent in 100,000 hr.

Table 2 records the creep tests upon an annealed hot-rolled steel bar with a chemical composition similar to the previous creep tests. The creep curves were published by Clark and Robinson (3). The temperature is 857 F (458 C) and some of the test bars have run 14,000 hr. In the first 700 hr, the creep curves had erratic fluctuations so that data from the first 1000-hr period cannot be used. Therefore, we can rely only upon the v_a and a constants found with longer-time data. Values in Table 2 were calculated from readings at 1000, 4000, and 10,000 hr. The b values indicate that the curves should be shifted up or down different amounts. The t_0 values are not given since the variation in the creep curves at the beginning of the test does not furnish data for the calculation.

This test shows that the creep curves for the longer periods follow the logarithmic law expressed by Equation [1]. The low values for the creep stress and high creep rates are due to the combined effect of anneal heat-treatment and to the banding in the steel structure.

A longitudinal section of the steel before it was tested is shown in Fig. 4. The white lines are ferrite containing nickel. They were originally dendrities segregated in the cast ingot, and during the rolling process were lengthened into threads running lengthwise in the rolled bar. To detect such structural irregularities

as are produced by the working of steel, all creep-test materials are studied by a longitudinal section and not a cross section of the bars. The white ferrite threads are weaker in creep strength and in strain hardening than the alloyed dark steel. The division of load changes with the creep extension, the white threads taking less load and the dark metal more load until a stable condition is reached. In this particular test with 43 per cent of cross-sectional area in white bands, it was estimated that the white and black metals began creep with equal stress loads but finally approached a condition where the white metal had about 0.3 and the black metal 1.7 times the average bar stress. It is only after such a structural and creep-strength equalization of load that stable conditions are obtained in creep. Depending upon the amount and actual form of banding present in the steel, one must expect distorted or even erratic creep-extension curves in the early part of the test.

Another example of changing conditions within the metal is found in steel castings with dendritic segregations insufficiently dispersed or broken up in the heat-treatment. Table 3 records the results of tests by Coffin and Swisher (2) on four items of 1.4 per cent manganese cast steel tested under constant loads at 842 F (450 C). Fig. 5 shows the metal structure in a section after 5450 hr in the creep test. The white metal is ferrite or nearly pure iron. The stronger dark alloyed steel encases the isolated volumes of iron. During creep extension, the dark metal mechanically squeezes and elongates the white volumes, producing a very slow but continuous strengthening of the total bar in creep. The same mechanical action in the beginning of the test gives considerable value to t_0, the stability indicator, and enlarges a the strain-hardening constant.

Levine (5) in the discussion of the paper by Coffin and Swisher (2), proposed an accurate formula to cover the particular test for which the values in Table 3 are given. This equation is similar to the one proposed earlier by Andrade (1). Neither formula covers the very long-time tests on metals of uniform structure. Due to the mechanical action on the encased dendrites described previously, the author believes it is impracticable to fit a formula to the creep curve for a dendritic casting because such a formula must contain a factor which would change with the amount and condition of the dendrites present in the steel.

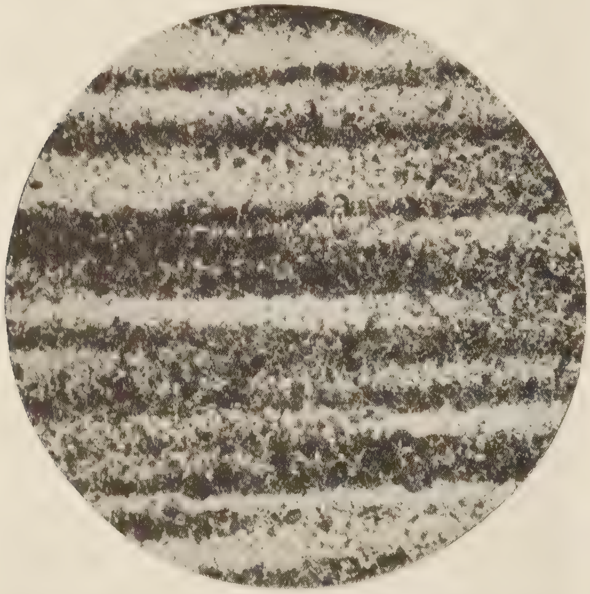

FIG. 4 STRUCTURE OF Ni-Cr-Mo STEEL BEFORE TESTING, SHOWING BANDING IN ANNEALED STEEL—×100

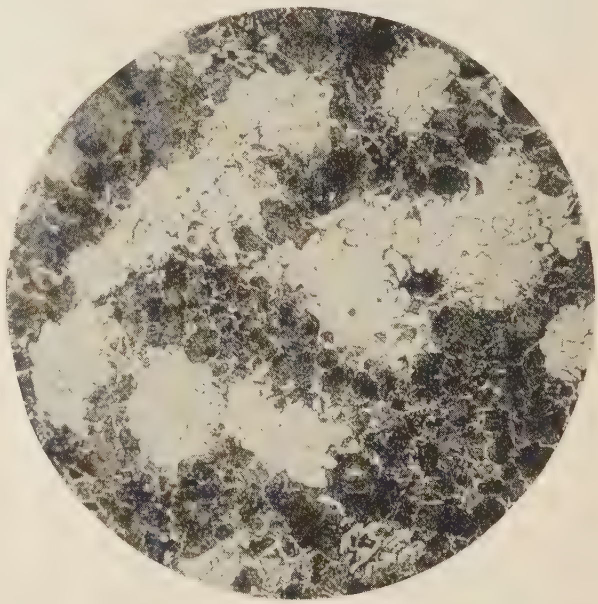

FIG. 5 STRUCTURE OF MANGANESE CAST STEEL AFTER CREEP TEST OF 5450 HR AT 842 F, SHOWING DENDRITES—×100

The creep characteristics and Equation [1] were checked by a number of other long-time tests on castings. Table 3 was selected as representative of the greatest difference for steel castings between calculated and test results. In all cases there was a con-

TABLE 3 CREEP-TEST DATA FOR MANGANESE CAST STEEL[a] AT 842 F (450 C)

Item no.	289	290	291	292
Test data:				
Stress, lb per sq in.	15000	12000	9000	6000
ϵ at 200 hr ($\times 10^{-3}$)	1.740	0.775	0.380	0.168
ϵ at 500 hr ($\times 10^{-3}$)	3.185	1.340	0.575	0.240
ϵ at 1000 hr ($\times 10^{-3}$)	5.075	2.030	0.778	0.307
ϵ_1 at t_1 hr ($\times 10^{-3}$)	18.400	6.100	1.840	0.600
t_1 = end of test, hr	5450	5450	5450	5450
Calculated data:				
v_a ($\times 10^{-7}$)	29.20	9.630	2.040	0.460
a ($\times 10^{-3}$)	1.43	0.695	0.337	0.146
b ($\times 10^{-3}$)	2.14	1.020	0.436	0.178
t_0	31.20	29.200	19.800	17.600
ϵ_1 at t_1 hr ($\times 10^{-3}$)	19.12	6.830	1.930	0.620

[a] Analysis: 1.42 Mn, and 0.31 C.

NOTE: Heat-treatment consisted of holding at 900 C for 6 hr and furnace cooling; holding at 850 C for 4 hr and furnace cooling; and holding at 750 C for 4 hr and furnace cooling.

NOTE: From a log-log graph at 842 F (450 C) with v_a = 1 per cent in 100,000 hr, the stress σ = 7600 lb per sq in., and $a = 0.25 \times 10^{-3}$.

TABLE 4 CREEP-TEST DATA AT 1022 F (550 C) FOR STAINLESS IRON ALLOYED WITH 3 PER CENT TUNGSTEN[a]

Item no.	298	299	300
Test data:			
Stress, lb per sq in.	12500	10000	7500
ϵ at 200 hr ($\times 10^{-3}$)	2.26	1.13	0.60
ϵ at 500 hr ($\times 10^{-3}$)	3.06	1.59	0.84
ϵ at 1000 hr ($\times 10^{-3}$)	3.80	2.08	1.10
ϵ_1 at t_1 hr ($\times 10^{-3}$)	5.73	3.86	1.95
t_1 = end of test, hr	10000	10000	10000
Calculated data:			
v_a ($\times 10^{-7}$)	10.10	5.220	2.880
a ($\times 10^{-3}$)	1.22	0.764	0.386
b ($\times 10^{-3}$)	0.76	0.733	0.348
t_0	4.20	9.100	8.000
ϵ_1 at t_1 hr ($\times 10^{-3}$)	14.20	7.540	4.080

[a] Analysis: 11.61 Cr, 3.01 W, 0.43 Mn, and 0.084 C.

NOTE: Heat-treatment consisted of furnace cooling from 800 C.

NOTE: From a log-log graph at 1022 F (550 C) with v_a = 1 per cent in 100,000 hr, the stress σ = 5600 lb per sq in., and $a = 0.2 \times 10^{-3}$. By test at 10,000 hr, σ = 12,000 lb per sq in.

tinued creep strengthening so that calculations based on test values would have a conservative error.

Another type of instability in metal structure is that of alloy segregations such as illustrated in "ferritic" or nonhardening steels. Table 4 records the creep-test and calculated constants for a constant-stress test at 1022 F (550 C) for 10,000 hr on a stainless iron with 3 per cent tungsten, annealed at 800 C. Using Equation [1] with the constants obtained from data in the first 1000 hr of test, the calculated total creep at 10,000 hr is 2 to 2.5 times the tested values and the creep rates are 13 to 25 times the actual, so there probably is a decided change in the metal structure continuing beyond the first 1000 hr of the creep test.

The micrographs of the steel before and after test show a continuous and uncompleted structural change in the steel, the total change increasing with the stress in each item and time in creep test. Fig. 6 shows the structure before test. Fig. 7 is item 298

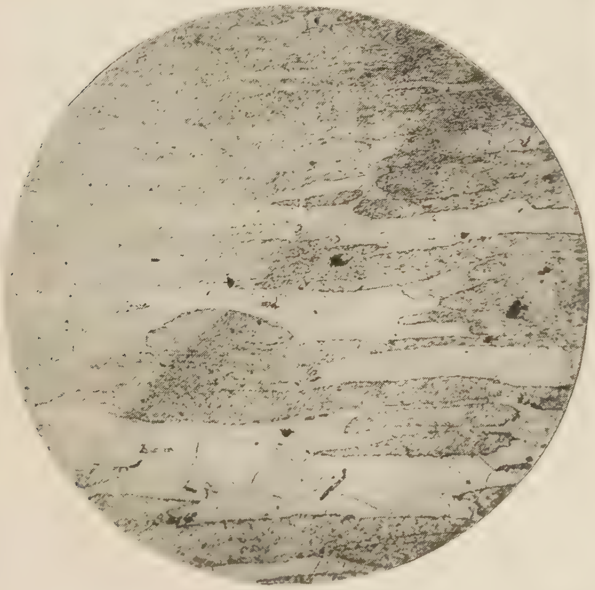

Fig. 6 Structure Before Testing of Stainless Iron Alloyed With 3 Per Cent Tungsten—×100

of Table 4 after 10,000 hr of creep test at 12,500 lb per sq in. The lower stressed items 299 and 300 of Table 4 are intermediate. The white streamers of chrome ferrite contain a progressive growth of a structural formation increasing with the stress or

TABLE 5 CREEP RATES OF ITEM 297

Time, hr	Stress, lb per sq in.	Temperature, F	Total creep
1000	15000	1022	7.50×10^{-3}
3400	step down	1022	0.47×10^{-3}
2350	step down	1112	0.65×10^{-3}

TABLE 6 PHYSICAL PROPERTIES BEFORE AND AFTER CREEP TEST OF STAINLESS IRON ALLOYED WITH 3 PER CENT TUNGSTEN

	Before creep test	After creep test			
Item no.	All	297	298	299	300
Tensile strength, lb per sq in.	99200	104200	94000	90000	90000
Elastic limit, lb per sq in.	59000	56000	56000	47000	47000
Elongation, per cent	25.5	23.5	26.5	26.0	26.0
Reduction of area, per cent	66.8	57.2	62.6	61.3	39.4
Charpy notch test results	58.0	22.5	31.1	35.0	36.6

total creep extension. That the structural changes were not completed at the end of the constant-stress test is evidenced by Fig. 8, where another test item 297 showed the successive creeps listed in Table 5. This test item shows an approach to a uniform structural formation within the steel. The physical properties at room temperature determined before and after creep test also indicate a metallurgical change as shown in Table 6. The tensile strength and elastic limit of items 299 and 300 had a de-

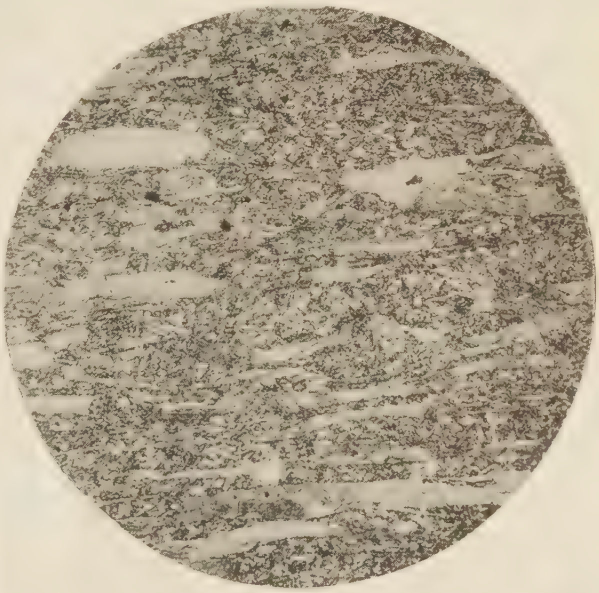

Fig. 7 Structure of Item 298 After Testing 10,000 Hr at 12,500 Lb Per Sq In. Constant Stress and 1022 F—×100

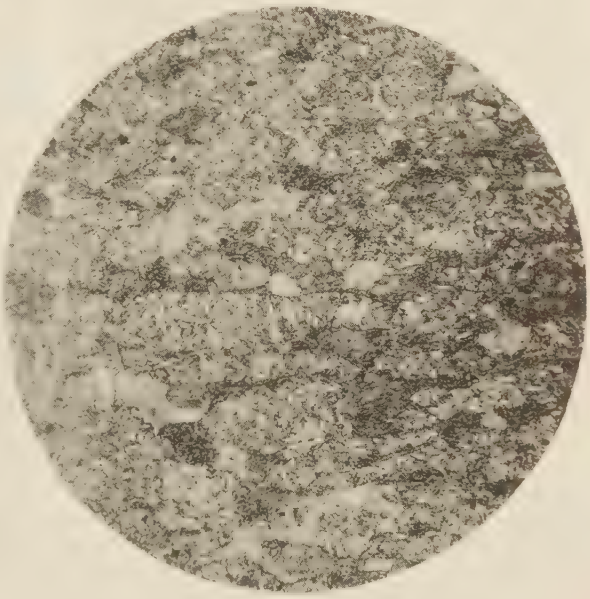

Fig. 8 Structure of Item 297 After Testing 4400 Hr at 1022 F and 2350 Hr at 1112 F—×100

cided drop. These items represent the first part of the structural change. Items 297 and 298 have increasing strength as the new structures become effective. The greatest change occurs in the Charpy notched impact values. The original impact was very high so that the values after creep test are not subnormal.

The creep strength of this material in the unstable condition

used in the creep test varies greatly as the structural changes progress. If the creep strength were stated as the stress that will produce a constant rate of creep of 1 per cent in 100,000 hr, then the creep strength of this stainless tungsten iron in the condition used for the 1022-F test, the results of which are given in Table 4, is 5600 lb per sq in. when based on Equation [1] and the creep curves during the first 1000 hr of test. Other results are 8100 lb per sq in. for the 4400-hr step-down test of item 297; 8600 lb per sq in. for the same item continued at 1112 F, a higher strength for a higher temperature; and 12,000 lb per sq in. from the actual creep-rate values in the constant-load test at 10,000 hr. It must not be assumed that uniform structure being approached in Figs. 7 and 8 is the strongest stable structure for creep. A heat-treatment at 950 C with slow cooling or oil quenching produces a homogeneous structure, while the oil quenching and tempering produces a creep strength of 14,500 lb per sq in. at 1022-F in a step-down test. Further studies on this material for structural strength and stability were discontinued in favor of another material.

The evidence of the structural changes in the material that occurred during the creep test of the stainless tungsten iron, results of which are given in Table 4, not only shows the difficulty of any theoretical creep formula to meet such changing conditions, but the improbability of predicting long-time performance of a nonhomogeneous or unstable material by means of any short-time or step-down test. It also plainly indicates the desirability in practice of using a steel that has an initial uniform structure which is in a stable condition.

The four long-time creep tests considered previously represent four of the principal types of instability in the structure of steel at elevated temperatures. The spheroidization of carbon is a slow-working and strength-decreasing metallurgical change that cannot be avoided in the steels containing carbon. The ferritic bands in rolled steel, the dendrites in steel castings insufficiently treated, and the alloy segregations as in the nonhardening steels are metal structures initially weak in creep and which usually but not always strengthen as the structure changes and creep extends. These nonuniform structures can be avoided in commercial products, and stable steels of higher creep strength can be obtained. The importance of the steel structure is emphasized by a comparison of Tables 1 and 2 where metallurgical conditions in the structure of the steel make a difference of four to one in the very long-time-tested creep strength.

Conclusions

With the assumption that initial creep at constant stress and temperature is a strain hardening phenomenon with a velocity that varies inversely with time, the creep extension due to this plastic flow is equal to a strain hardening constant multiplied by the logarithm of time. In the viscosity of solids, under stable conditions there would be a viscous flow at a constant rate which would produce a creep extension of constant creep rate multiplied by time. The total creep extension for a structurally stable metal at any time is equal to the sum of these two products. The creep properties of a steel under constant stress and temperature, including the initial creep, is defined by (1) the strain-hardening constant and (2) the constant-creep rate.

An unstable steel is one in which a slow structural change occurs within the metal during creep. For such material a third constant must be added to measure the degree of instability, all the effects being represented by Equation [1]. This third constant or remainder in the creep formula should be small for a stable metal.

The application to a creep test for 1000 hr consists in plotting creep extension against log of time so that the bend in the creep curve approaches a straight line. Creep extensions at 200, 500, and 1000 hr are read from the curve and the three creep constants calculated.

Many of the 1000-hr constant-stress tests were followed by a step-down test a total of 3000 hr. This served as a check on constants calculated from data in the first 1000 hr, and micrographs of structures before and after test, with the specimens mounted and prepared together, checked the stability constant of the material. Routine calculations were applied to 72 creep tests of 1000 to 2000 hr duration. The asymptotic constant creep rate was always a little less than the tested rate at 1000 hr, and the strain-hardening constant for the initial creep could be judged closely by the total creep at 1000 hr. The practical importance of the formula lies in the sensitivity of the stability-of-metal constant which can detect small continuous changes in the metallurgical structure of the metal, changes which were difficult to find in the physical properties or micrographic studies.

The accuracy of the formula was also checked by 39 very long-time tests extending from 5000 hr to 5 yr. From data in the first 1000 hr, the creep curves were projected to end of test. When calculations and test results did not agree the stability constant always had an appreciable value (or $b > 0.5a$). It was then necessary to prove the structural instability of the steel.

From the longer creep tests, four sets of curves were selected and presented in detail as representative of four principal types of metallurgical instability.

Spheroidization of carbon is illustrated by four tests at 842 F (450 C) which agree with the creep formula for $2^1/_2$ yr, then began a slow increase in creep until at the end of 5 yr the total creep in the four bars was 6 to 9 per cent greater than calculated values. Spheroidization at this temperature is such a slow-working factor that it cannot be detected by the stability constant and does not enter the creep calculations until after $2^1/_2$ yr. It becomes of increasing importance with higher temperatures. The tests on this steel indicate a creep strength of 47,000 lb per sq in. for an asymptotic constant-creep rate of 1 per cent in 100,000 hr at 842 F (450 C).

Ferritic banding in rolled steel of approximately the same chemical analysis as the previous test is represented by five items tested up to 14,000 hr. The banded threads of ferrite caused erratic creep readings for 700 hr. From 1000 to 14,000 hr, the test and creep formula agree. The tested creep stress of 10,800 lb per sq in. produces an asymptotic constant-creep rate of 1 per cent in 100,000 hr at 857 F (458 C). With 43 per cent of the total bar section covered by the area of ferrite threads, which if assumed at zero strength, the creep strength would be 19,000 lb per sq in. Hence, the difference in creep strength between the later value and the previous test must be accounted for by other metallurgical conditions of the steel.

Dendrites in steel castings insufficiently broken up by heat-treatment are illustrated by a 5450-hr test on four cast-steel bars at 842 F (450 C). This and other tests on cast material in similar condition always gives from the data in the first 1000 hr an increased strain hardening constant and an excessive stability constant. The creep strength slowly increases as extension progresses and at 5450 hr the tested total creep is 7 per cent below the calculated value.

Alloy segregations producing an unstable steel in creep is represented by four items from the same bar of 12 per cent chromium iron with 3 per cent tungsten tested for 10,000 hr at 1022 F (550 C). The segregations of chrome ferrite in clear streamer form progressively broke up during creep, then acicular formations began in the interior of each section until the structure could not be recognized as being related to the original. The stability constant in the first 1000 hr signaled a changeable steel. The same data indicated a creep stress of 5600 lb per sq in. while tests at 10,000 hr gave a creep stress of 12,000 lb per sq in.

for a creep rate of 1 per cent in 100,000 hr. By test the total creep was 50 per cent of the calculated value.

There are other types of metallurgical instability in the creep of steels but the four referred to in this paper are believed to lead in importance. The tests were selected as representing unstable creep material that can be found in high-quality commercial products but which can be placed in a stable and stronger creep condition.

Appendix 1

When a bar of metal previously unstrained is held at a certain constant temperature and a load is applied producing a certain constant stress, three distinct movements occur: (1) The bar elongates elastically, instantly responding to the strain according to Young's modulus; (2) the bar extends plastically, beginning with a relatively high velocity which quickly decreases with time; and (3) the bar continuously elongates at a smaller constant rate which is viscous flow in accordance with the theory of solid viscosity.

These three phenomena exist at the same time. As elastic elongation is independent of time it need not be further considered. Assume that the rate of plastic extension v' varies inversely with time t', then the product $v't'$ equals a constant. The initial velocity of plastic flow cannot be infinite, but must begin at a high value v_0 which corresponds to time t_0. This is expressed as

$$v't' = k = v_0t_0$$

and

$$t' = t + t_0$$

where t is the actual time measure. Then

$$v' = \frac{v_0t_0}{t + t_0}$$

Let v_a equal the constant viscous-flow rate. Then the flow rate v at any time t or creep rate for constant stress and temperature is

$$v = v_a + \frac{v_0t_0}{t + t_0}$$

and the total creep extension at any time is

$$\epsilon = \int_{t_0}^{t} v dt = v_a t + v_0t_0 \log (t + t_0) - v_0t_0 \log t_0$$

These equations can be written in simpler form for application to creep tests by using common logarithms and

$$a = 2.3v_0t_0$$

$$b = a \log_{10} t_0 \ldots\ldots [2]$$

Practically, $\log_{10} (t + t_0) = \log_{10} t$. Then, for constant-stress creep tests

$$v = v_a + \frac{a}{2.3t}$$

and

$$\epsilon = v_a t + a \log_{10} t - b \ldots\ldots [1]$$

where ϵ = total creep elongation, in. per in. of length; v_a = constant creep rate for constant stress and temperature; a = strain-hardening constant; t = time, hr; v = creep rate at any time, in. per in. per hr; b = test constant for a particular test and material; and t_0 = time modulus for rate of strain hardening, or a test constant indicating stability of metal structure.

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2 "Flow of Steels at Elevated Temperatures," by F. P. Coffin and T. H. Swisher, Trans. A.S.M.E., vol. 54, 1932, paper APM-54-6, p. 59.

3 "An Automatic Creep-Test Furnace," by P. H. Clark and E. L. Robinson, *Metals & Alloys*, vol. 6, February, 1935, p. 46.

4 "Testing of Materials for Service in High-Temperature Steam Plants," by R. W. Bailey and A. M. Roberts, Proceedings of the Institution of Mechanical Engineers, vol. 122, 1932, pp. 209–284.

5 Discussion by L. Levine of "Flow of Steels at Elevated Temperatures," by F. P. Coffin and T. H. Swisher, Trans. A.S.M.E., vol. 54, 1932, paper APM-54-6, p. 66.

Indexes to A.S.M.E. Papers and Publications

THE following pages will serve as a guide to papers and publications of the A.S.M.E. during the calendar year 1936, and also to publications developed by the technical committees. The publications of the Society may be classified in two general groups, regular and special, and were as follows:

REGULAR SOCIETY PUBLICATIONS, 1936

Mechanical Engineering, monthly (see index on pages RI-99–106)
A.S.M.E. Transactions, monthly (see index on pages RI-107–115)
Mechanical Catalog, 1936–1937 edition.

SPECIAL PUBLICATIONS ISSUED IN 1936

1935 Oil Engine Power Cost Report

Heat Transfer (Papers contributed by the Heat-Transfer Committee of the Process Division of the A.S.M.E. at its Annual Meetings in December, 1933 and 1934)

Marketing Research (A Selection of Books and Articles on the Purpose, Scope, and Techniques of Marketing Research, by George W. Kelsey, with the cooperation of Mary L. Alexander and William F. Turnbull)

Standards

Brass Fittings for Flared Copper Tubes, approved January, 1936, published January, 1936

Socket Set Screws and Socket Head Cap Screws, approved February, 1936, published February, 1936

Pipe Plugs, approved October, 1936, published October, 1936

Circular and Dovetail Forming Tool Blanks, approved November, 1936, published November, 1936

Chucks and Chuck Jaws, approved November, 1936, published November, 1936

Lathe Spindle Noses, approved November, 1936, published December, 1936

Power Test Codes

Stationary Steam-Generating Units (revision), approved January, 1930, published January, 1936

Instruments and Apparatus

- Part 2, Pressure Measurement; Chapter 2, Static and Total Pressure, Static Holes and Tubes, Impact Tubes, and Chapter 3, Pipes for Pressure Measurement, approved, May, 1934, published October, 1936
- Part 3, Temperature Measurement, Chapter 2, Radiation Pyrometers, approved July, 1935, published May, 1936
- Part 10, Flue and Exhaust Gas Analyses, approved April, 1934, published June, 1936
- Part 14, Linear Measurements, approved January, 1934, published May, 1936
- Part 20, Smoke-Density Determinations, approved October, 1933, published May, 1936

Papers Presented at A.S.M.E. Meetings, 1936

The complete technical programs of the meetings of the Society and of its Professional Divisions have been published in *Mechanical Engineering* and may be located by consulting the index on pages RI-99–106. A considerable number of papers and reports included in these programs were not published during the year in either Transactions or *Mechanical Engineering*, but were issued in mimeographed or photo-offset form. Complete sets of these are on file for reference purposes at the office of the Society and the Engineering Societies Library, under the title of "Miscellaneous Papers Presented at A.S.M.E. Meetings, 1936." Photostatic copies of any of the papers may be secured from the Library at regular rates. A list of these papers and reports follows.

Miscellaneous Papers Presented at A.S.M.E. Meetings, 1936

Reed, Robert F., Lithographic Research
Reports of Committees to A.S.M.E. Council
Rosenzweig, S., Vibration Control of Internal-Combustion Engines
Schwarz, Ralph H., and Winkler, J. Homer, Heat-Treated Electrotypes
Scott, Howard, The Problem of Quenching Media for the Hardening of Steel
Seider, E. N., Application of Fouling Factors in the Design of Heat Exchangers
Smith, C. R., Air Line Operation
Smith, V. Weaver, Modern Steam Drilling Equipment
Stone, M., Recording Extensometer for Rolling Mills
Thirlwall, J. C., Application of Diesel-Electric Locomotives in the Buffalo Area
Tillinghast, Harold M., Color Printing in Daily Newspapers
Weber, E. R., Diesel Engines in Railroad Service
White, A. E., and Good, C. W., Research Activities at the University of Michigan
Wilson, G. C., Engine Governors

Publications Developed by the Technical Committees

The Society's technical committees, the first of which was organized many years ago and all of which have been continuously at work on codes, standards, research, and other special reports, have developed a series of publications of permanent value to the membership. The following list is first presented here for record and for ready reference. This list covers the entire group of publications of these committees completed to date which are now available.

To assist the members in securing copies of these publications the sale price is also given. It should be recalled, however, that a discount of 20 per cent is allowed to members of the A.S.M.E. on all pamphlet publications, except standards and the cases specially noted.

RESEARCH

Dynamic Loads on Gear Teeth (1932), $1.50
Determination of Carbonate, Phosphate and Hydroxides in Boiler Waters (1933), $1.75
Fluid Meters
  Part 1—Theory and Application (under revision) $2.50
  Part 2—Description of Meters (1931), $1.75
  Part 3—Selection and Installation (1933), $1.50
Tests on Electrical Equipment for Drilling Rotary Drilled Oil Wells (1933), $0.85
Tests on Steam Equipment for Drilling Rotary Drilled Oil Wells (1932), $0.85
Roll-Neck Bearings (1935), $1.50
Bibliography on Cutting of Metals (1866–1930), $1.25
Bibliography on Deterioration of Condensing Equipment (1845–1930), $1.25
Bibliography on Effect of Temperature Upon Properties of Metals (1828–1931), $1.25
Bibliography on Management Literature (1931), $1.75
Supplement to Management Bibliography (in preparation)
Bibliography on Mechanical Springs (1678–1927), $1.25
Bibliography on Woods of the World (1928), $1.25
Steam Tables and Mollier Diagram (1930), $2.50

POWER TEST CODES

Test Codes for

Atmospheric Water-Cooling Equipment (1930), $0.45
Centrifugal and Rotary Pumps (1927), $0.50
Compressors and Exhausters (1934), $0.95
Condensing Apparatus (1925), $0.55
Displacement Compressors and Blowers (1925), $0.50
Evaporating Apparatus (1925), $0.50
Feedwater Heaters (1925), $0.35
Gas Producers (1928), $0.55
Internal-Combustion Engines (1929), $0.55
Liquid Fuels (1930), $0.35
Reciprocating Steam Engines (1935), $0.65
Reciprocating Steam-Driven Displacement Pumps (1925), $0.65
Refrigerating Systems (1926), $0.55
Solid Fuels (1931), $0.55
Speed-Responsive Governors (1927), $0.45
Stationary Steam-Generating Units (1930), $0.60
Steam Locomotives (1926), $0.55
Steam Turbines (1928), $0.55

Supplementary Codes and Publications

Definitions and Values (1931), $0.40
General Instructions (1929), $0.35
Instruments and Apparatus—
  Part 1—General Consideration (1935), $0.35
  Part 2—Pressure Measurement, Chapter 1, Barometers, and Chapter 6, Tables, Multipliers and Standards (1929), $0.45
  Part 2—Pressure Measurement; Chapter 2, Static and Total Pressure, Static Holes and Tubes, Impact Tubes, and Chapter 3, Pipes for Pressure Measurement (1936), $0.65
  Part 3—Temperature Measurement, Chapter 1, General; Chapter 5, Pyrometric Cones; Chapter 6, Liquids-in-Glass Thermometers; and Chapter 7, Bourdon Tube Thermometers (1931), $0.75
  Part 3—Temperature Measurement; Chapter 2, Radiation Pyrometers (1936), $0.55
  Part 3—Temperature Measurement, Chapter 8, Optical Pyrometers (1933), $0.40
  Part 4—Head Measuring Apparatus (1933), $0.35
  Part 6—Electrical Measurements (1934), $1.25
  Part 9—Heat of Combustion (1932), $0.40
  Part 10—Flue and Exhaust Gas Analyses (1936), $1.35
  Part 11—Determination of Quality of Steam (1931), $0.45
  Part 12—Measurement of Time (1932), $0.35
  Part 13—Speed Measurements (1930), $0.45
  Part 14—Linear Measurements (1936), $0.55
  Part 16—Density Determinations (1931), $0.35
  Part 17—Determination of the Viscosity of Liquids (1931), $0.50
  Part 18—Humidity Determinations (1932), $0.45
  Part 20—Smoke-Density Determinations (1936), $0.65
  Part 21—Condenser Leakage Tests. Chapter 1 (1928), $0.45
  Part 21—Leakage Measurement, Chapter 2, Boiler and Piping; Chapter 3, Steam Engine Leakage (1932), $0.40

BOILER CODE

Power Boiler Code (1935) Including Materials Specifications, with 1936 Addenda, $2.50
Locomotive Boiler Code (1935) with 1936 Addenda, $0.55
Low-Pressure Heating Boiler Code (1935) with 1936 Addenda, $0.65
Miniature Boiler Code (1935) with 1936 Addenda, $0.50
Suggested Rules for Care of Power Boilers (1935), $0.70
A.S.M.E. Unfired Pressure Vessel Code (1935) with 1936 Addenda, $0.75
A.S.M.E. Boiler Construction Code, Combined Edition (1935) with 1936 Addenda, $5.50
Boiler Code Interpretation Service Data Sheets, $0.15
  Annual Subscription, $2.50

STANDARDS

Machine Shop Practice Standards

Shafting and Stock Keys (B17.1—1934), $0.45
Code for Design of Transmission Shafting (B17c—1927), $0.75
Woodruff Keys, Keyslots, and Cutters (B17f—1930), $0.35
Tolerances, Allowances, and Gages for Metal Fits (B4a—1925), $0.50
American Standard Screw Threads for Bolts, Machine Screws, Nuts, and Threaded Parts (B1.1—1935), $0.60
Wrench-Head Bolts and Nuts and Wrench Openings (B18.2—1933), $0.50
Slotted Head Proportions; Machine Screws, Cap Screws, and Wood Screws (B18c—1930), $0.45
Track Bolts and Nuts (B18d—1930), $0.40
Round Unslotted Head Bolts, Carriage, Step, and Machine Bolts (B18e—1928), $0.40
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T-Slots, Their Bolts, Nuts, Tongues, and Cutters (B5a—1927), $0.35
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Milling Cutters: Nomenclature, Diameters, Thickness, and Other Important Dimensions (B5c—1930), $0.75
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Pipe and Fittings

Wrought-Iron and Wrought-Steel Pipe (B36.10—1935), $0.50
Code for Pressure Piping (no discount) (B31—1935), $1.00
Cast-Iron Pipe Flanges and Flanged Fittings for 25 Lb Maximum Saturated Steam Pressure (B16b2—1931), $0.40
Cast-Iron Pipe Flanges and Flanged Fittings for 125 Lb Maximum Saturated Steam Pressure (B16a—1928), $0.50
Cast-Iron Pipe Flanges and Flanged Fittings for 250 Lb Maximum Saturated Steam Pressure (B16b—1928), $0.50
Cast-Iron Pipe Flanges and Flanged Fittings for 800 Lb Maximum Hydraulic Pressure (B16B1—1931), $0.35
Malleable-Iron Screwed Fittings for 150 Lb Maximum Saturated Steam Pressure (B16c—1927), $0.40
Cast-Iron Screwed Fittings for 125 and 250 Lb Maximum Saturated Steam Pressure (B16d—1927), $0.35
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Miscellaneous Standards

Engineering and Scientific Charts for Lantern Slides (Z15.1—1932), $0.50
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Hose-Coupling Screw Threads for $^1/_2$, $^5/_8$, $^3/_4$, 1, $1^1/_4$, $1^1/_2$, and 2 in. Connections (B33.1—1935), $0.25
Rolled Threads for Screw Shells for Electric Sockets and Lamp Bases (C44—1931), $0.35

SAFETY

Safety Code for Elevators (A17—1931) (no discount), $1.00
Safety Code for Mechanical Power-Transmission Apparatus (no discount) (B15—1927), $0.35

JOINT CODE

API-ASME Code for Unfired Pressure Vessels for Petroleum Liquids and Gases (1936), $1.00 (no discount)

Index to Mechanical Engineering

Volume 58, January–December, 1936

(BR) denotes Book Review, (C) Correspondence, (Ed) Editorial, (S) Survey of Engineering Progress

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Index to A.S.M.E. Transactions

Volume 58, 1936

The A.S.M.E. Transactions for 1936 was issued monthly, four of the twelve issues being the *Journal of Applied Mechanics*, the page numbers for which are preceded by the letter A.

Each paper in the other eight issues is given a letter symbol (appearing at the top of the page) which identifies the Division or committee sponsoring it. The key to the symbols follows: AER (Aeronautics), FSP (Fuels and Steam Power), HYD (Hydraulics), MSP (Machine-Shop Practice), MAN (Management), MH (Materials Handling), OGP (Oil and Gas Power), PME (Petroleum Mechanical Engineering), PRO (Process), RR (Railroad), RP (Research), and TEX (Textiles). These symbols are accompanied in each case by the volume number of Transactions and the number of the paper; i.e., HYD-58-11 indicates the eleventh paper sponsored by the Hydraulic Division to be published in volume 58 of Transactions.

The Record and Index for the year 1936 was issued as supplements, one in February and one in August, 1936, and one in January, 1937. The page numbers for these supplements are designated by the symbol RI.

(D) as used in this index denotes a discussion of a paper; (BR) book review.

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1936 ANNUAL MEETING PAPERS IN THIS ISSUE

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T. Baumeister, Jr.

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